International Marine Engineering

DECEMBER, 1906.

THE ENGLISH BATTLESHIP DREADNOUGHT.

Greater interest has been manifested in the design and construction of the battleship which we illustrate herewith than has been the case with any previous naval construction in many years. This is not only due to the great celerity with which the ship was built, but also, and more markedly, to the extremely high military qualities which she possesses.

The *Dreadnought* has a displacement of 17,900 tons, the length between the perpendiculars being 490 feet, while a beam of 82 feet makes her the broadest war vessel afloat. She is fitted with four screws operated at a maximum of 300 revolutions per minute by turbine engines of the Parsons type, designed to develop 23,000 horsepower. Steam is furnished by eighteen Babcock & Wilcox boilers, working under a pressure of 250 pounds per square inch, and fitted for operation with either coal or oil fuel. Each boiler has twenty elements. The total grate surface is 1,560 square feet, and heating surface 55,400 square feet, the ratio being 35.5 to 1. The boilers are arranged in three stoke holds and the products of combustion are discharged through two funnels, the after one being of very large size.

STERN VIEW OF THE DREADNOUGHT, SHOWING THE IMMENSE BEAM OF THE SHIP.
(Photograph, Cribb, Southsea.)

The trials of this ship were very satisfactory, there having been a considerable series for the purpose of learning all that was possible to ascertain about the operation of the machinery. A thirty-hour trial on October 3 and 4 at about one-fifth power showed a total horsepower for the four turbines averaging 5,013, with a corresponding speed of ship of 13 knots. The steam pressure in the boilers was 220 pounds; at the engines, 210 pounds. The revolutions of the inner shafts averaged 205 per minute; of the wing shafts, 182 per minute. The power developed on the inner shafts averaged 1,709 for each shaft, while the outer shafts showed an average of 797 horsepower each. The coal required per horsepower per hour figured out at 2.6 pounds.

On October 4 progressive trials at various powers were run, lasting in each case from one to two hours and varying in the power developed from 1,303 to 15,875 horsepower. The steam used per horsepower hour was shown to decrease from 58.5 pounds at the lower power to 16.6 pounds at the higher.

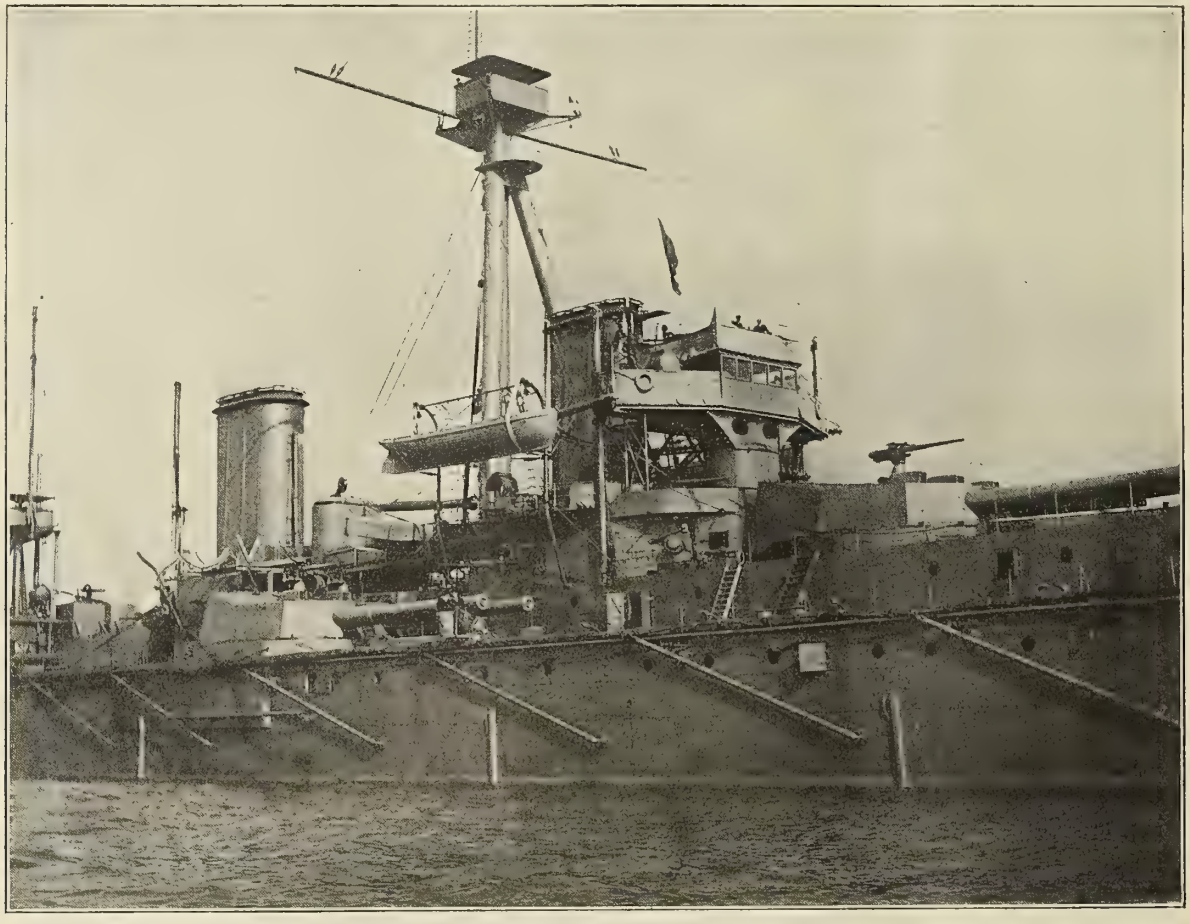
Trials over the measured mile at seven-tenths power showed a total of 17,698 horsepower, with a corresponding speed of 19.42

knots, this having been attained with the shafts operating at an average of 298 revolutions per minute. A thirty-hours' trial at this same power with a pressure of 232 pounds on the boilers showed a total of 16,930 horsepower, a speed of 19.3 knots, a fuel consumption of 1.7 pounds per horsepower hour, and a mean of 291 revolutions per minute.

Full power trials were run October 9, 1906, that over the measured mile having shown an average of 27,518 horsepower with 21.6 knots, and with an average of 337 revolutions per minute. The eight-hours' full power trial took place this same day, the steam pressure in the boilers being 241 pounds. The speed of 21.25 knots was developed with an average of 24,712 horsepower, the propellers operating at a mean of 329 revolutions per minute. The coal burned per indicated horsepower per hour figured out at 1.51 pounds, the steam having been 15.56 pounds, and water evaporated per pound of coal 10.03 pounds. The slip of the propellers at full speed was found to be 26 percent, and at 19½ knots 21 percent. The horsepower of the engines was measured directly by means of torsion meters of the Denny & Johnson type, which measure the angular distortion of the shaft itself.

The battery, which is the most powerful ever yet placed upon any ship of war, consists in the main of ten 12-inch guns mounted in pairs in five turrets, as shown in the illustrations. Three of these turrets are on the center line, one being forward on the forecastle deck, one aft on the quarter deck, and one aft of the funnels, but forward of the after turret, and on the same level with it. The other two turrets are mounted, one upon either beam, well forward of the waist, and serve to give the tremendous bow fire accruing from six guns of such extreme power. The broadside fire, as is seen, is from eight of these guns, which is the same as that given the two new American battleships *Michigan* and *South Carolina*, though it is stated that the English ship has guns of higher power.

With her great length and great size, this ship would be able to maintain her speed in a seaway to a much higher degree than is the case with most vessels against which she would have to contend. She is the equal in speed of most of the fast armored cruisers belonging to her rivals, and, of course, of overwhelmingly superior offensive power. Not only is her battery immensely superior to that of any other battleship afloat, but her extremely high speed will enable her to choose such a fighting distance as would best suit her own peculiar qualities, and crush an enemy with a concentrated fire of eight heavy guns, against which no other battleship afloat could oppose more than four, and these in the main of inferior individual power.



VIEW SHOWING DETAILS OF THE EMPLACEMENT OF TURRETS AND CONTROL STATIONS. NOTE THE TRIPOD MAST.
(Photograph, Cribbs, Southsea.)

Among the many innovations introduced into the design and construction of this ship, that one most apparent to the observer is the form of the forward mast, which is a tripod. This obviates all necessity for stays, and renders the mast much less liable to be destroyed by a single heavy shell than is the case with previous ships. Another novelty is in the location of officers' quarters forward, with the crew aft. Maneuvering power has been largely aided by the fitting under water of a double stern and two rudders. The deadwood is completely cut away, and the rudders, which are quite 20 feet apart, are of the balanced type. This reduces materially the strain on the steering gear, which is of the Napier Brothers type.

Much praise has been bestowed upon the plant of Babcock & Wilcox boilers, constructed by the builders of the machinery, Messrs. Vickers Sons and Maxim, Limited, Elswick, England. On the trials nearly 17 horsepower was developed per square foot of grate area, and maintained for lengthened periods.

BOW OF THE DREADNOUGHT DURING ANCHOR TRIALS, SHOWING THE TOWERING FORECASTLE.
(Photograph, Cribb, Southsea.)

FLOATING DRY-DOCK CONSTRUCTION.

BY WILLIAM T. DONNELLY.

(*Concluded.*)

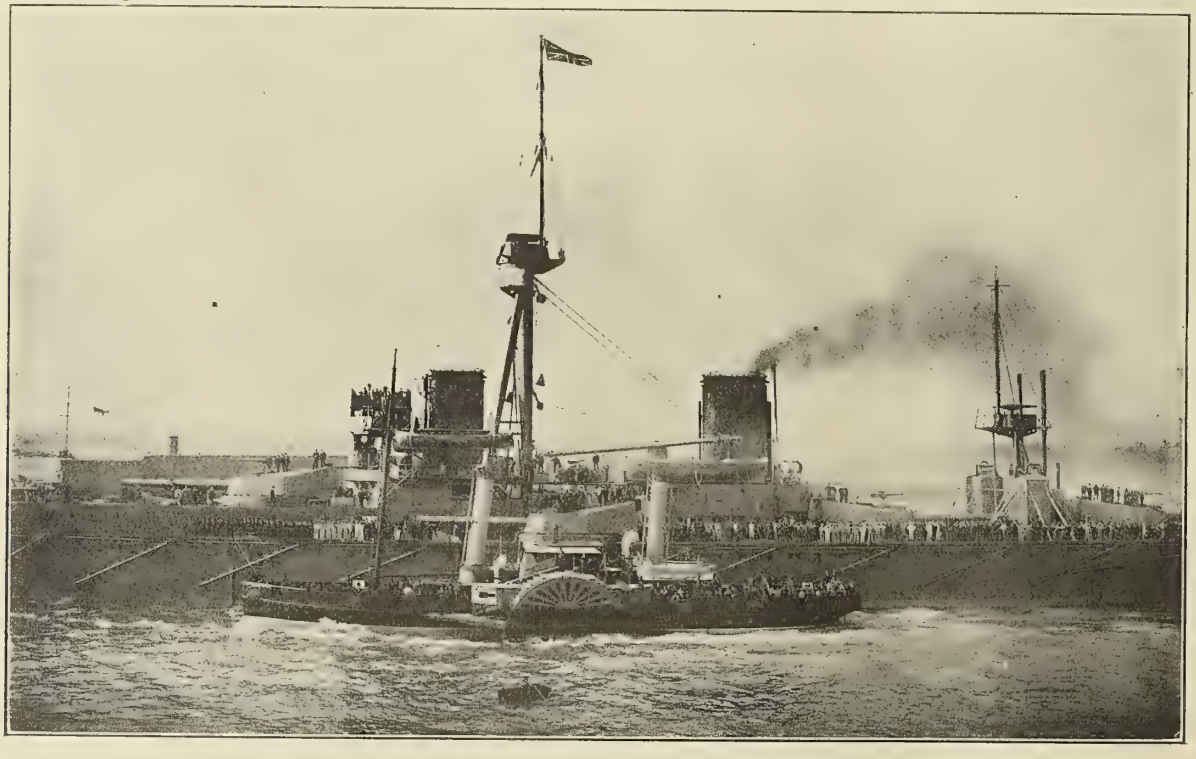
OPERATION OF FLOATING DOCKS.

We will now take up the operation of floating dry docks along lines which will be of more particular interest to those who are already familiar with the subject.

With the first form of dry dock of which we have any record in this country, as previously explained, if it can be grounded, or rest upon the bottom when being pumped out, and if it does not float until it is practically empty, it will operate all right, but if any considerable amount of water is present at the time it floats, one end will certainly be slightly heavier than the other, consequently this end will sag down, and the water will flow towards the lower end, causing it to still further settle, and any considerable angle of inclination would be liable to cause the displacement of the vessel upon the blocks.

There was a boiler and engine and a group of pumps at the center of each side of the dock. The pumps all drew their water from a central well or compartment. This compartment was connected to the various other divisions of the dock by gates. The general plan of pumping was to start all the pumps and lower the level of the water in the pumping compartment, and then to open the gates from the end compartment and allow the water to flow by gravity to the pumping compartment. This, you will readily understand, would work out properly if the dock was always on a perfectly level keel, but if for any reason one end became lower than the other, the delivery of water from that end would not be as rapid, and the subsequent pumping would tend to increase the difficulty, and the only method of correcting this condition would be to stop pumping, close all gates and admit water from the outside of the dock to the high end or side, and then commence pumping over again.

It was soon discovered that the only possible way to handle this dock was to place the vessel, whatever her length, in the center of the dock, and then to pump all compartments at the same rate, stopping to correct any tendency to roll as soon as it became apparent. The continual docking of vessels shorter than the total length of the dock has resulted in straining the dock to such an extent that the center of the dock, longitudinally, is now about 4 feet lower than the ends. The dimensions of this dock, 300 feet long by 100 feet wide, in a single structure, are such that it has never been possible to entirely remove it from the water, and it is certainly remarkable that after more than 50 years it is still possible to dock vessels upon it.



PARTIAL SIDE VIEW OF DREADNOUGHT, SHOWING LOCATION OF HEAVY GUNS.
(Photograph, Cribb, Southsea.)

The old balance dock now in use in the Erie Basin, is, I think, the oldest dock now in use in the harbor. As originally built, it had gates at each end similar to those referred to in previous section on the Adamson dock, from which it was undoubtedly an outgrowth, and, so far as I am able to learn, the builders believed that they would be able to operate it entirely as a floating structure without resting upon the bottom at any time.

The operation of this dock was intended to be as follows: The gates at one or both ends were opened, and water was admitted into the body of the dock, and it was sunk low enough for the vessel to be moved in; after the vessel was in place the gates were closed at both ends and pumping commenced, the idea being to remove the water between the side walls and end gates, but, as there were no divisions either longitudinally or laterally, to confine the water, as soon as pumping was commenced the water would flow to the lowest end or side, and it was found impossible to keep the dock and vessel in a horizontal position. The gates were then abandoned and the attempt made to operate the dock as a box or balance dock, and in this connection I wish to call your attention especially to the arrangement of the pumping machinery.

In the next development, which is the small sectional dock, commonly so called, the sections were made small enough to be controlled in sinking by locking logs connecting them; when a vessel is in place the vessel itself serves as the connection between the sections to steady and maintain them in correct relative position. The sections of this type of dock were made about 20 feet in the direction corresponding to the length of the dock, and of a width proportionate to the length of the dock, so as to give working room on each side of the largest vessel the dock would take. The machinery was placed on one side of the dock,

with a water connection or leader extending through the center bulkhead to the other side.

As this was the first type of floating dock to be extensively used, and as it embodies most of the principles that are in use in the more recent and larger docks, I will try and explain the method of pumping and handling somewhat in detail. It was a dock of five sections, which was quite a common number; these represent a dock, each section of which is about 20 feet long by about 60 feet wide. You will understand that we speak of the shorter dimension of the section (20 feet) as the length, because it is that dimension which forms part of the longitudinal dimension of the dock as a whole. Each section was divided at a point corresponding to the center line of the dock by a watertight bulkhead. At the extreme outside of each section there was a flood gate.

The pumping machinery was located on one side of the dock, and as it was necessary to bring the water from the other side beyond the center bulkhead, a leader or pipe was provided, the flow through which was controlled by a gate or valve similar to the flood gates. The pumping machinery consisted of four single acting pumps for each section, and were operated through rocker arms, connecting rods and shafting. To allow for movement between sections, the connection of the power from one section to the next was made by a combined double universal and slip joint. This mechanism permitted a considerable movement in any direction without interfering with the transmission of the power.

To limit the movement of the sections, there was provided on each side, at the junction of the wings with the deck, a continuous locking log, which passed through strong keepers near each end of each section. These logs were not intended to take any of the strain of lifting the vessel, but were to control and keep in alinement the sections while being lowered, and until the vessel was firmly resting upon the blocks, when it will be readily understood that the vessel herself would make a firm and strong bond between the sections. Power was furnished for pumping by a boiler and engine located on the middle section.

The operation of the dock is about as follows. The keel and bilge blocks having been prepared for the vessel to be docked, all the flood gates are opened about half way and the entering water causes the dock to settle; any tendency for one section or part of the dock to descend faster than the other is corrected by slightly closing the corresponding gate. When the deck of the dock approaches the surface the gates are gradually closed, and, as the water floods the dock between the wings, the different sections and the dock as a whole are carefully leveled. This is quite important, to insure complete control of the dock. When the deck is entirely submerged, the flood gates are opened wider and the dock descends to the necessary depth to allow the vessel to enter over the blocks. At this depth all valves are closed, and the vessel is moved and correctly located over the blocking and secured by lines at bow and stern to the sides or wings of the dock. The pumping machinery is then operated, slowly at first, to bring the keel blocks gently against the keel of the vessel without disturbing them. As soon as the dock is pressed firmly against the bottom of the vessel the bilge blocks are drawn under the bilges. This may be done while the pumping is going on slowly, or, if the vessel is large, the pumping is stopped while the blocks are drawn. When this is accomplished, the pumping machinery is speeded up and the water removed more rapidly.

The dockmaster in the meantime watches the dock and vessel, and if for any cause there is a tendency for one side or end of the dock to rise faster than the other, he does not change the rate of pumping, but causes water to be let into the section or side which is coming up faster than the other. Of course, if the vessel is not as long as the dock, or does not rest evenly upon all the sections, those not supporting the vessel will naturally rise faster than those that are to support the vessel, and from the beginning of the pumping it will be necessary to check their rise by admitting water.

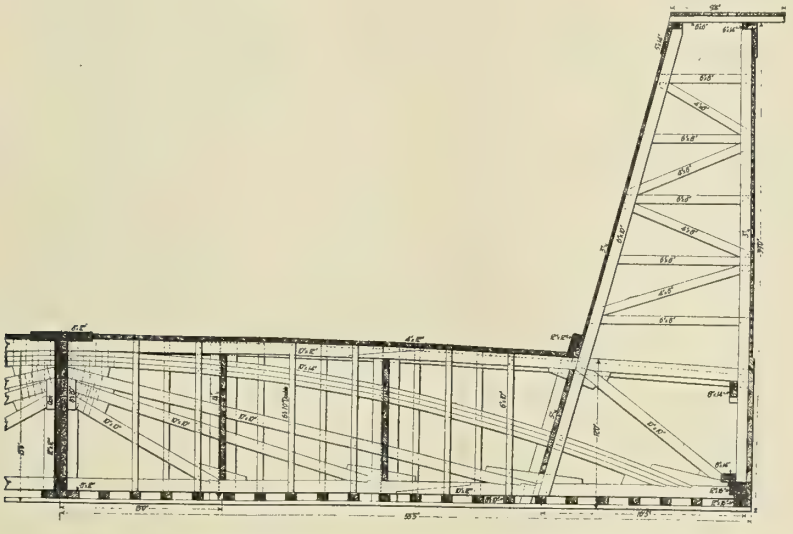


FIG. 7.—TRANSVERSE SECTION OF WOODEN DRY-DOCK, SHOWING TRUSS.

When the pumping commenced, the level of the water in the interior dock was approximately that upon the outside, and consequently the pumping machinery would be required to deliver the water only at a slight elevation above that from which it was received, and the load upon the engine at this time is small, but at this time the water is being removed from the wings only, and as the area or bulk of these wings (exclusive of the pump wells, which do not assist, as they are full of water at all times while the dock is submerged) is small comparatively to the area or bulk of the vessel at the water line, the vessel rises slowly, while the difference between the inside and outside level of the water in the dock changes rapidly, thus increasing the load upon the pumping plant. This will continue until the interior level of the water (that in the interior of the dock) has been lowered down to the level of the deck. At this time, such a dock as the one referred to, with a vessel of the full capacity of the dock upon it, drawing when afloat, say, 12 feet, will have been lifted 3 feet, and the interior water level, which at the start was about 2 feet below the exterior, has been increased to a difference of 12 feet.

When the interior water level reaches the level of the deck of the dock, a very marked change takes place in the conditions. The area to be pumped at once increases to the entire area of

the dock and is many times more than the corresponding area of the vessel, which, in fact, is rapidly decreasing. The necessary consequence is that the vessel and dock from this time on rise out of the water much faster than the reduction of the interior water level, and the head against which the pumps are required to deliver the water rapidly falls with a corresponding decrease of the load on the engine. If we consider this dock as having a depth in the body of 8 feet and that at the time the deck of the dock reaches the water there is 1 foot of water remaining in the dock, then the head against which the pumps will be delivering at this time will be 7 feet, a decrease of almost one-half from the maximum head of 12 feet.

With the deck of the dock at the surface of the water another marked change takes place. If the vessel being raised has occupied only four sections, the fifth section will now have in it about 7 feet of water, or the flood gates of this section have been so regulated that the interior water level is only 1 foot below the exterior, but from this point, or, as we say, when the water leaves the deck of the dock, it is necessary to pump all sections, both loaded and unloaded alike.

The lack of comprehension and grasp of the principles of hydraulics as applicable to these structures has resulted in a general mistrust which is not warranted. With the deck of the dock flush with the surface of the water, I would like to call your attention to some of the considerations of stability of dock and vessel.

As you are well aware, a vessel when afloat without cargo (which is the usual case when dry docking is to be done) is in a condition of minimum stability, and this stability will rapidly disappear as the vessel is raised from the water. Therefore it is necessary in dry docking a vessel in a floating dry dock to make some provision for the stability of dock and vessel as they rise together out of the water. The stability of the vessel relative to the dock is provided for by the bilge blocks, which are drawn under the bilges (corresponding to the widest part of the bottom of the ship) before she is lifted any considerable distance, but as soon as this is done the dry dock must assume the double responsibility of caring for itself and the vessel also. The wings of a floating dry dock are often considered as serving only to carry the pumping machinery, but this is far from the fact. Their most important use is to furnish stability for the vessel and dock as they rise out of the water.

The details of this problem are too intricate and involved to be considered at this time, but if you will consider the fact that stability in its last analysis depends upon turning moment applied to the floating mass, you will at once see that the wings of a floating dock, situated at a considerable distance from the center line of the dock and vessel considered as a single floating structure, are admirably placed to counteract the loss of stability of the vessel as she rises out of the water. Further than that I would state that the proportions are such in a properly designed dock that the combined stability of dock and vessel are at all times more than the stability of the vessel when afloat.

From the consideration of this dock we will pass to the next development, or the "box or balanced dock," as illustrated by Fig. 3. As has been previously stated, this dock is the type which is the most popular at this time for small and moderate sizes, or from 500 to 3,000 tons capacity.

The special features of construction of this dock are a single rigid structure, a center longitudinal watertight bulkhead, and a number of watertight transverse bulkheads dividing the dock into a sufficient number of separate compartments so that the controllable pumping may control the action of the dock. The machinery of these docks is located on one side only, and the water from the opposite side is secured by leaders and control valves in the same manner as described for the sectional docks. The flooding and sinking of the dock is also carried out in the same manner, care being taken in sinking to have the water come over the deck at each end slowly, and as evenly as possible. The vessel, when smaller than the capacity of the dock, should always be located near one end, and the pumping regulated by the flood gates so as to bring up the dock on an even keel, both transversely and longitudinally.

As to the particular construction of one of these docks, a cross-section of a recently designed dock of about 1,500 tons capacity, shows a width of 68 feet and the depth at the center corresponding to the longitudinal center line is 10 feet 8 inches, and the entire depth, including the wing, is 32 feet 11 inches. The distribution and dimensions of the timber need little explanation, a second longitudinal bulkhead being added on each side of the center to give longitudinal rigidity. There are twenty pumps, located in three pump wells on one side of the dock. On the opposite side are three similar pump wells, but without pumps. The object of these pump wells is to equalize the buoyancy on each side of the dock, as the pump well space does not assist in lifting. The construction of the prow or outrigger on each end shows an opening in its center, to allow for the lowering of the rudder of a vessel when on the dock.

We will now pass to the most recent type of floating dry dock, illustrating the largest development in the commercial floating dock in this country up to the present time. (See Fig. 4.)

This shows a balanced sectional dry dock of five sections, having an over all length of 468 feet, a width of 110 feet 10 inches, and a lifting capacity of 10,000 tons. The arrangement of sections, locking logs, bulkheads and pumping machinery is clearly shown. There are thirty pumps on each side, and the manner in which they are operated is quite clearly shown.

This type of dock is the invention of Mr. Frederic C. Lang, and patents were issued to him under dates of April 10 and August 21, 1900. As we have followed the developments of these structures, it will seem to most of you that this dock is so much a development out of the previous structures that its production called for little invention, but this view is opposed to the most positive evidence to the effect that there was a real and well-known demand for a large floating dock in this port for more than twenty years. In fact, the two Dodge-Burgess docks, of 3,000 and 3,500 tons, previously referred to, were the largest in existence in this country, and no actual advance had been made in fifty years.

Much of the unqualified success of the undertaking, even from a 10,000-ton floating dry dock was first brought to the attention of the engineering firm with which I was connected (Faber du Faur & Donnelly), and I became convinced that in connection with the development of an entirely new plan of construction an increase of size of about three times over any previous wooden floating dry dock was to be undertaken, you will readily understand that the situation called for serious thinking, particularly as the next largest work in this line was about fifty years old.

Much of the unqualified success of the undertaking, even from an engineering point of view, is due to the inventor, Mr. Lang, and his long experience in the building and operating of the different types of smaller docks; from years of experience in handling them as a dockmaster he was in possession of a knowledge of all the conditions which such a structure must meet. During the designing, one of the first features which received consideration was the relative advantage of wood and steel construction. After a careful investigation, somewhat to my surprise, I am free to admit, wood won out upon its merits, and when the reduction of first cost is also made an element, the comparison becomes impossible, as the cost of a steel dock of this size is almost, if not quite, twice as much as one of wood. As to durability, it would seem as if little could be asked in that direction, with the evidence before us of the three largest floating docks in the port, each of which is about fifty years of age. Some one may ask if the original wood of these docks has not been replaced, perhaps several times, in that period. To this I would answer that the most careful investigation on my part has failed to discover any evidence of any such change of timber in

the body of the dock, or the trusses which support the weight of the ship. In the upper work, which has been exposed to the sun and weather, there has been considerable replacing of material.

To protect these wooden structures below water from the attacks of worms and the teredo, it is the practice to treat the bottom and sides with a coat of coal tar, then sheath them with hair felt saturated with creosote, and this with a covering of 1-inch rough boards treated with creosote and arsenic. A dock prepared in this manner may remain in the water almost indefinitely.

The length of the trusses was 110 feet and the depth 12 feet 2 inches. The upper and lower members of the truss passed through the center bulkhead, and the longer diagonal braces met in the center. The shorter braces abutted against the center bulkhead on opposite sides. The center bulkhead was watertight, as previously described; on each side there were three additional bulkheads to stiffen the dock longitudinally. These were not made watertight. The members of the truss were proportioned for a maximum strain of 1,000 pounds per square inch, tension and compression.

The distribution of the strains in a dock of this kind are somewhat peculiar. The first pumping removes water only from the wings located at each end of the trusses, and, with the vessel resting only on the keel blocks at the center of the dock, we have a beam loaded at the center and the force applied at the ends.

When the vessel is entirely raised, the wings not only do not support any of the burden, but themselves act as a reverse load upon the trusses and the weight of the ship is almost entirely borne by that portion of the trusses between the wings. Careful investigation of these conditions develops the curious fact that under some conditions of use the greater strain upon the trusses will be produced by a part of the vessel's weight acting through the buoyancy of the wings upon the ends of the trusses.

The pumps were arranged in groups as shown in Fig. 4. The pump well has a watertight floor at about the level of the deck of the dock, and a spillway is provided through the outer wall. The pumps are square wooden boxes extending through the floor of the pump well, and to within a few inches of the bottom of the dock. Near the bottom of the pump barrel, which is 16 inches square, is the foot or suction valve. About half way up the pump well there is a rocker arm, to each end of which is attached a pump rod and to one end of which the connecting rod is attached, and delivers the motion from the line shaft above; the stroke of the pumps is 20 inches, and as they are single acting, they each deliver about 2 cubic feet of water per revolution. They operate at from 60 to 75 revolutions per minute.

The pumping machinery on each side of the dock is operated by a vertical slide valve engine having a cylinder 20 inches in diameter by 24 inches stroke. There was no attempt in the power equipment to obtain high economy, as at the most they would operate but one or two hours a day. There are two upright boilers on each side, having a diameter of 72 inches and a height of 13 feet, built for a working pressure of 125 pounds per square inch. A pressure of from 80 to 90 pounds is found to be all that is required to operate the dock.

The construction of floating docks, and their development up to this time, have been almost entirely in the hands of practical men, and the details of their construction and equipment have been quite thoroughly worked out, but the hydraulic principles involved and the structural strains involved in their use have received little attention, and are not generally understood.

ABSTRACTS FROM DISCUSSION.

All drift-bolts were of galvanized iron; on the extreme upper work, which is always above the water and can be frequently painted, black screw-bolts were used. In the design, particularly the parts below the water, the study was to use as few screw-bolts as possible. The only long screw-bolts are 1¼ inches diameter through the main trusses alongside of the longitudinal bulkheads; these were galvanized. When metal such as drift-bolts is entirely in wood it will not deteriorate, but when used as through bolts it will rust in time, even when galvanized.

Nearly all iron vessels carry a docking plan and generally have a straight keel, but wooden vessels are sometimes very much out of shape. We have docked a side-wheel steamer used as a towboat on the Hudson river, the keel of which was 2½ feet out of a straight line. This boat was built straight and had sagged to a double curve, the keel being down 2½ feet in the center below the highest place, which was about two-thirds of the way from the center to the end. The ends dropped a foot below the highest point. In blocking for a vessel such as this we use patent keel blocks, that is, blocks composed of wedges that can be drawn in such a way as to be adjusted while the vessel is in place in the dock. Every fourth block is usually made a patent or adjustable block, and when docking a vessel, the shape of which is in doubt, we arrange the blocking in a straight line, and then as soon as the vessel touches the blocks we haul on the patent blocks. If the keel is straight, none of the blocks can be moved, as they will all be in touch with the bottom. If it is not straight, all will be hauled until they touch the keel. Then the vessel will be taken out of the dock and it will be pumped up, when the shape of the keel will be clearly shown by the patent blocks, and the intermediate blocks can all be built up to correspond, after which the vessel can be docked in the usual way.

The object of the arch form of truss is to get a good hold of the lower truss head; if you carry a straight member out to that point, you will get very little actual connections coming in at such an acute angle, but by bringing the arch over we get a very good connection at that point, which takes the load off the wings when the dock is up; when there is nothing on but the ballast that is used to sink the dock. In the 10,000-ton dock we have to put in 500 tons of ballast to each section, or 2,500 tons of ballast. When the dock is pumped up we have then a great bulk of stone piled in each wing and the dock is supported by a uniform pressure of water across the entire bottom. The center of that force will be at the center line; we have the weight of the structure and 500 tons on each side tending to break the dock in two in a reverse direction.

I have considered the one-piece dock and it seems to me that if you undertake to make a structure 800 or 1,000 feet long, and endeavor to keep the depth of the bottom down to a reasonable figure, say 20 feet, the proportion of length to depth in the truss is such as to call for an enormous cross-section of metal, if you are to keep the deflection down to a reasonable figure. I am of the opinion that the problem can be satisfactorily solved in all its requirements by a sectional dock.

Speaking of a commercial dock, you must come right down to the question whether it will pay or not; it isn't a matter of the largest dock in New York harbor. This dock has been a commercial success simply because it is built small enough. It will dock a vessel 525 feet long; that doesn't go into the largest class, of which there are only a few vessels. As the vessel gets larger it has more compartments; those vessels can hobble over to the other side, where they have basin docks. A very few exceedingly large docks will do for the largest vessels, and it is hardly reasonable for us to make them here, if they have them on the other side. The question is, if you have your large dock here, will the vessel stay here? Not unless she absolutely has to.

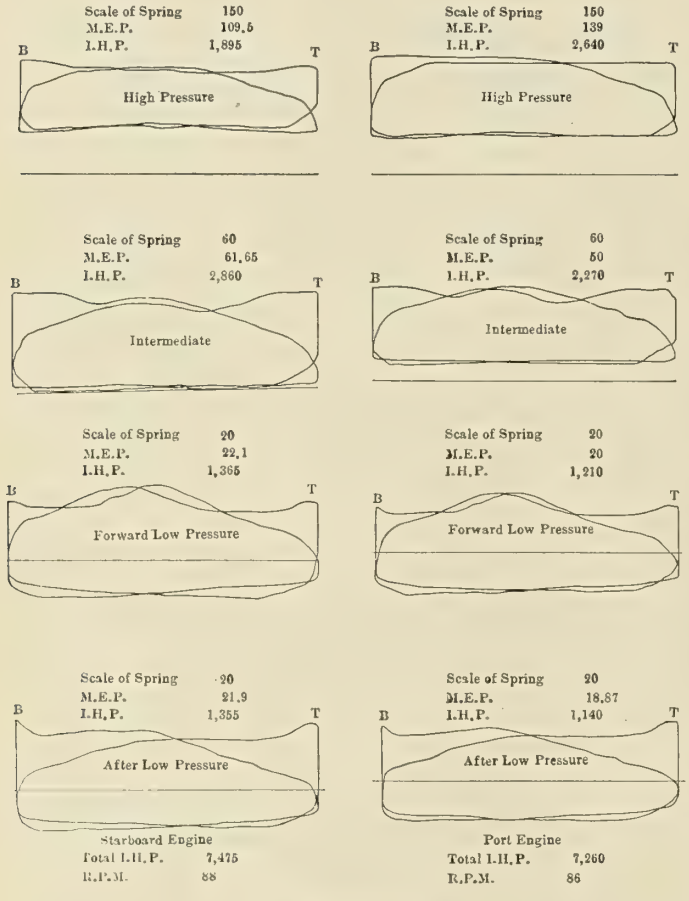
The Mallory Line of coastwise steamships, plying between New York and Brunswick, Ga., Key West, Mobile and Galveston, has been sold to Charles W. Morse for a consideration said to be $7,500,000. The ships of the line are *San Jacinto,* 6,069 tons; *Denver,* 4,579; *Concho,* 3,724; *Sabine* and *Nueces,* 3,350 each; *Comal, Lampassas* and *Alamo,* 2,940 each; *San Marcos* and *Colorado,* 2,800 each, and *Rio Grande,* 2,566. A new steamer is now being built at Newport News, and two others are to be added.

DOCK TRIAL OF THE UNITED STATES BATTLESHIP MINNESOTA.

BY H. B. GREGORY.

The United States battleship *Minnesota*, the fifth 16,000-ton batteship for the navy, was given her dock trial at the yard of her builders, the Newport News Shipbuilding and Dry Dock Company, August 14 to 17 inclusive.

Each engine was run separately on two consecutive days. The trial started with the port engine on August 14 and 15, followed by the starboard engine on the 16th and 17th. The first day's run, in each case, was of about eleven hours' duration. Four boilers were used, and the maximum revolutions attained were 68.7 for the port and 66 for the starboard engine. On the second day's run of each engine, the power developed was largely in-



SAMPLE INDICATOR CARDS.

creased. Six boilers were in operation, and the maximum revolutions reached 86 and 89, respectively, port and starboard. The port engine was in operation about ten and one-half hours on this day, while the starboard engine on the corresponding day was run only about five and one-half hours, thus making the total running time about twenty-one and one-half hours and sixteen and one-half hours respectively.

The boilers used each day were so chosen that each one was in operation for at least one day's run, and in this way each unit of the boiler plant received a practical test. The firing was systematic, the periods of firing and raking being regulated by electric signal operating automatically at equal intervals. The blowers were run throughout the trial at an average of about 350 revolutions, showing a slight fraction of an inch air pressure, but the fire rooms were not strictly closed.

Everything in the machinery plant worked admirably, and the results of the trial were most pleasing and satisfactory in every way, giving good grounds to expect successful speed results, when the official preliminary trial is run.

The following are the principal dimensions of the vessel and machinery, and the trial data:

HULL DIMENSIONS.

Length between perpendiculars	450 feet 0 inches
Length over all	456 feet 4 inches
Beam, extreme	76 feet 10 inches
Draft, normal	24 feet 6 inches
Displacement, normal	15,970 tons
Designed speed at normal displacement	18 knots

ENGINE DATA.

Type, four-cylinder triple expansion:	
Number	2
H. P. cylinder, diameter	32½ inches
I. P. cylinder, diameter	53 inches
L. P. cylinders (two), diameter, each	61 inches
Stroke	48 inches
Designed indicated horsepower	16,500
Designed steam pressure at high-pressure cylinder	250
Revolutions per minute, designed	120

BOILER DATA.

Type	Babcock & Wilcox
Number	12
Heating surface, square feet	52,752
Grate surface, square feet	1,100
Ratio, G. S. to H. S.	1 to 47.96
Steam pressure	265

SYNOPSIS OF TRIAL.

	Port.	Starb'd.
Steam pressure at boilers, gage	260	250
Steam pressure at engine, gage	255	240
Steam pressure at H. P. steam chest, gage	245	240
Steam pressure at 1st receiver, absolute	90	98
Steam pressure at 2d receiver, absolute	20	22
Vacuum in inches	23	26
Revolutions per minute	86	88
Mean effective pressure, H. P. Cyl.	139	109.5
Mean effective pressure, I. P. Cyl.	50	61.65
Mean effective pressure, F. L. P. Cyl.	20	22.1
Mean effective pressure, A. L. P. Cyl.	18.87	21.9
Mean effective pressure, aggregate referred to low-pressure cylinders	59.5	59.9
Indicated horsepower, H. P. Cyl.	2,640	1,895
Indicated horsepower, I. P. Cyl.	2,270	2,860
Indicated horsepower, F. L. P. Cyl.	1,210	1,365
Indicated horsepower, A. L. P. Cyl.	1,140	1,355
Indicated horsepower, total	7,260	7,475
Indicated horsepower, total both engines	14,735	

TEMPERATURES, F°.

Engine room, working platform	87 to 128
Engine room, lower grating	99 to 109
Engine room, upper grating	103 to 130
Fire room, working level	102 to 141
Fire room, grating	96 to 135
Fire room, at blowers	121 to 150
Main injection	78 to 93
Outboard delivery	109 to 127
Feed tank	91 to 120
Main feed	174 to 204

Benjamin H. Warren, of the firm of Kafer, Mattice & Warren, 60 Wall street, New York City, died suddenly October 19. Mr. Warren was graduated from the Naval Academy in 1874, and after four years of service entered commercial life, his work culminating with the presidency of the Allis-Chalmers Company. He is the second of the new firm above mentioned to die during the short time which that firm has been in existence, Mr. Kafer having passed away a few weeks after the firm was organized.

CLEARANCE AND RECEIVER DROP IN THE DESIGN OF COMPOUND ENGINES.

BY P. F. WALKER.

It seems to be the general practice of writers on the subject of engine designing when dealing with compound engines, to omit consideration of clearance altogether and to deal with drop or pressure between cylinders in a manner more or less indirect or vague. For accurate designing, and especially for a correct determination of the point of cutoff in the low-pressure cylinder, it is necessary to take both of these conditions into account. In the numerical calculations to follow in this paper, the writer has not striven particularly for the best proportions, nor does he expect that all losses have been corrected in the best possible degree of accuracy, but the object has been to outline a rational method of dealing directly with the quantities mentioned. The question of distribution of the work between the cylinders has not been considered at all, but in the case cited the common proportion assumed between the piston areas provides for a nearly equal division.

The method of dealing with drop between cylinders will be readily understood from the regular course of the discussion, but the treatment of clearance may need some preliminary explanation. The following is the common discussion as given in steam engine texts.

Considering the theoretical indicator card of a simple engine, as in Fig. 1, let

V_1 = the volume swept through by the piston during its stroke up to the point of cutoff.

V_2 = the entire volume of piston displacement.

V_c = the volume of the clearance space.

P_1 = absolute pressure of the entering steam.

P_2 = absolute pressure of steam at the end of expansion.

P_b = absolute back pressure of steam.

Then $V_1 + V_c$ = total volume of steam in the cylinder at cutoff.

$V_2 + V_c$ = total volume of steam at end of stroke.

and $\dfrac{V_2 + V_c}{V_1 + V_c}$ = the true ratio of expansion.

This ratio is seen to be less than $\dfrac{V_2}{V_1}$, which is the reciprocal of the fractional portion of the stroke traversed by the piston at cutoff, and which is the ratio of expansion when clearance is neglected.

If the expansion be assumed to be according to the rectangular hyperbola, so that PV = constant, the value of P_2 will be found from the equation

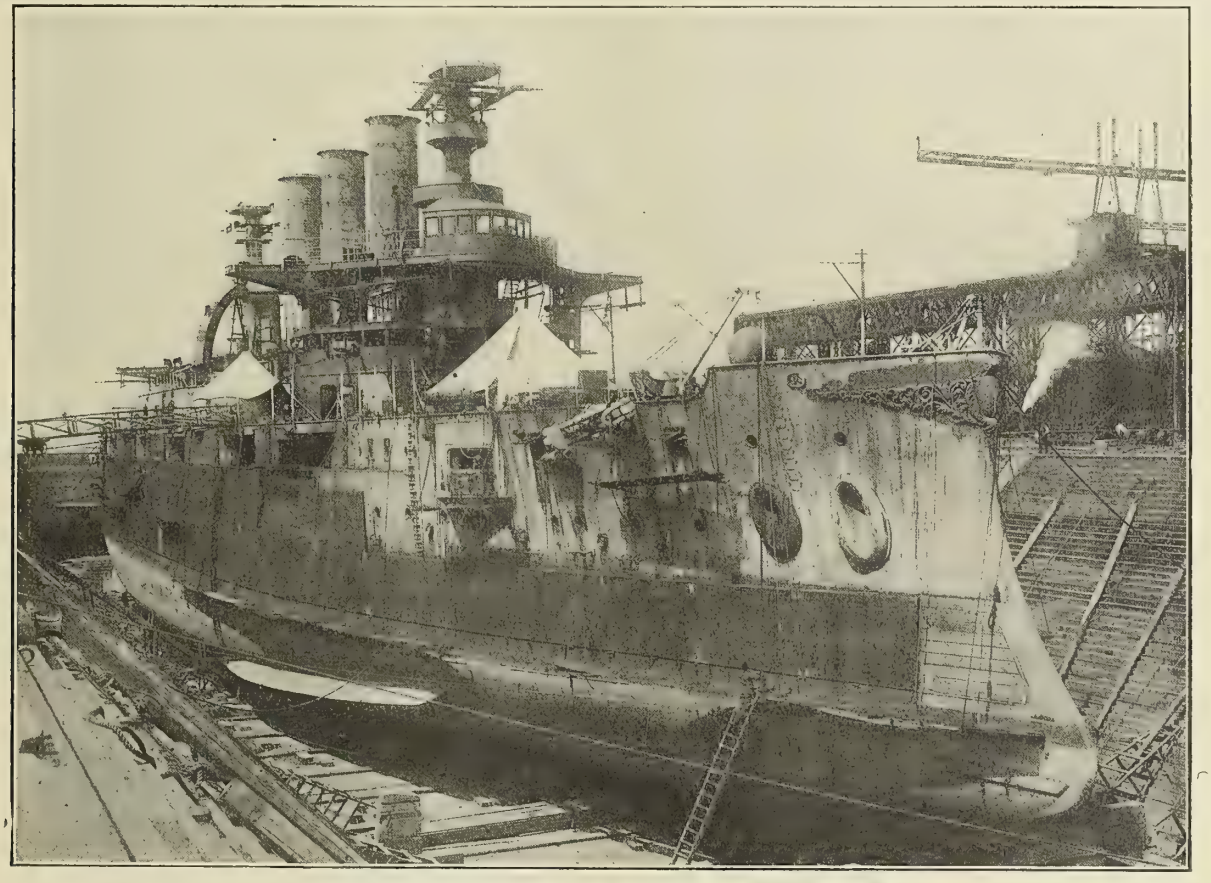
$$P_1 (V_1 + V_c) = P_2 (V_2 + V_c),$$

from which

$$P_2 = P_1 \frac{V_1 + V_c}{V_2 + V_c}$$

This value of P_2 is greater than $P_1 \times \dfrac{V_1}{V_2}$ which is the expression that results when clearance is neglected, and hence it follows that the presence of a clearance volume serves to raise the expansion line, and thus to increase the power of the engine. That this increase of power is accompanied by increased wastes of heat energy does not concern us in the present discussion. The main point is that if the engine is designed for a certain power without regard to clearance, proper allowances being made for the reduction of mean effective pressure due to losses from cylinder condensation, wire-drawing, etc., it will, when constructed with the necessary clearance spaces, develop more than the rated power; and the fact is to be recognized that it is as much a fault to produce engines of too great power as of too small power.

The work done during one stroke of the piston is represented on the indicator card by the area *a b c e f d a*. Work is the product of a force times a distance, and on the indicator card the vertical dimension represents the force and the horizontal dimension the distance. As actually drawn, the vertical dimension



THE BATTLESHIP MINNESOTA IN DRY DOCK AT NEWPORT NEWS.

is in terms of pounds pressure per square inch, which may be expressed in terms of total force on the piston by multiplying by the area of the piston; while the horizontal dimension represents the length of stroke on a reduced scale. If, however, the factor of piston area be multiplied into the length of stroke, the result is the volume of piston displacement, and the final product of force times distance thus reduces to the product of pressure per unit area times the volume through which that pressure acts. For convenience, the pressures will be kept in terms of pounds per square inch, which means that 144 times the true volume in cubic feet will be taken for the horizontal dimension, this being the result of multiplying piston area in square inches by length of stroke in feet. In engine designing, however, the cylinder dimensions are unknown quantities, and the mean effective pressure is the only quantity involved on the assumed indicator card, so that any convenient values may be assigned to the terms V_1 and V_2, so long as they give the true ratio of expansion that has been selected as the proper one for the conditions. The mean effective pressure will then be found by dividing the effective area of the card by the effective length, V_2.

Since pressures are best measured from the zero line, the effective area of the card is best found from the combination of the three portions, as follows:

Effective area $= (a\ b\ c\ e\ f\ d\ a) = (g\ b\ c\ h) + (h\ c\ e\ k) - (a\ f\ k\ g)$.

It is readily shown by the principles of the calculus that the area under a curve whose equation is $PV =$ constant, between volume limits of V_a and V_b, pressures being measured from the zero or vacuum line, is,—

$$\text{Area} = PV \times \text{hyp. log.} \frac{V_b}{V_a}$$

We have then the following expressions for the three portions of the card above mentioned,—

$$\text{Area } (g\ b\ c\ h) = P_1 \times V_1$$
$$\text{Area } (h\ c\ e\ k) = P_1 (V_1 + V_c) \times \text{hyp. log.} \frac{V_2 + V_c}{V_1 + V_c}$$
$$\text{Area } (a\ f\ k\ g) = P_b V_2.$$

From these there results as the value of the mean effective pressure on the theoretical card,

$$\text{m. e. p.} = \frac{P_1 V_1 + P_1 (V_1 + V_c) \times \text{hyp. log.} \frac{V_2 + V_c}{V_1 + V_c} - P_b V_2}{V_2} \quad (1)$$

This is the true mean effective pressure for the perfect engine, and will be thus referred to later.

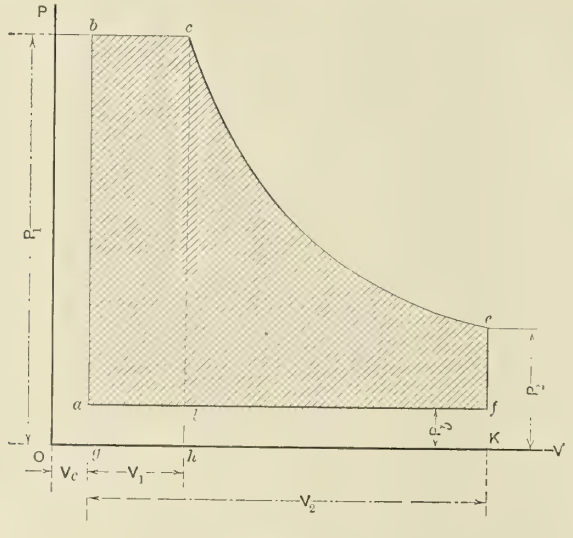


FIG. 1.

When the clearance volume is neglected altogether, this equation reduces to the form,

$$\text{m. e. p.} = \frac{P_1 V_1 + P_1 V_1 \times \text{hyp. log.} \frac{V_2}{V_1} - P_b V_2}{V_2}$$
$$= P_1 \left(\frac{V_1}{V_2} + \frac{V_1}{V_2} \times \text{hyp. log.} \frac{V_2}{V_1}\right) - P_b$$

But $\frac{V_2}{V_1} =$ ratio of expansion $= r$, when clearance is neglected.

$$\text{Hence m. e. p.} = P_1 \left(\frac{1 + \text{hyp. log. } r}{r}\right) - P_b \quad \ldots\ldots\ldots\ldots (2)$$

This is the form of equation ordinarily used in engine designing, the ratio of expansion being considered simply as the reciprocal of the fraction of stroke traversed by the piston at cutoff.

Another expression that is sometimes used results from considering clearance in the same manner as in equation (1), but the m. e. p. is calculated as though the piston swept through the clearance volume in addition to the piston displacement proper. Equation (1) then becomes,

$$\text{m. e. p.} = \frac{P_1 (V_1 + V_c) + P_1 (V_1 + V_c) \times \text{hyp. log.} \frac{V_2 + V_c}{V_1 + V_c} - P_b (V_2 + V_c)}{V_2 + V_c}$$

Since $\frac{V_2 + V_c}{V_1 + V_c}$ is in this case the ratio of expansion, r, this becomes,

$$\text{m. e. p.} = P_1 \left(\frac{1 + \text{hyp. log. } r}{r}\right) - P_b \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

This is identical in form with equation (2), but r here represents the true ratio of expansion. The error involved in assuming the longer stroke of piston is considerable, however, giving too great a value to the mean effective pressure.

A numerical example will show the differences between these three equations clearly. Take the case of a steam engine using steam at 125 pounds per square inch absolute pressure, cutting off at one-fourth stroke, with back pressure 18 pounds per square inch, and with 5 percent clearance. Since the actual size of the engine is independent of the pressures, we may for present purposes put V_1 equal to unity and V_2 equal to 4, while V_c equals 5 percent of 4, or 0.2 cubic foot.

$$\text{From (1), m. e. p.} = \frac{125 \times 1 + 125\ (1 + .2) \times \text{hyp. log.} \frac{4.2}{1.2}}{4} - 18 \times 4$$
$$= 60.22 \text{ pounds per square inch.}$$

This is the true theoretical mean effective pressure.

$$\text{From (2), m. e. p.} = 125 \left(\frac{1 + \log. 4}{4}\right) - 18.$$
$$= 56.58 \text{ pounds per square inch.}$$

$$\text{From (3), m. e. p.} = 125 \left(\frac{1 + \log. \frac{4.2}{1.2}}{\left(\frac{4.2}{1.2}\right)}\right) - 18$$
$$= 62.47 \text{ pounds per square inch.}$$

To disregard clearance in designing is here seen to result in a theoretical mean effective pressure too small by 6.1 percent, and to consider it only to the extent of determining the true ratio of expansion without changing the form of the expression for mean effective pressure to that in equation (1), makes that quantity too large by 3.7 percent. By doing the small amount of additional work required in equation (1), a value is found that has been corrected for one of the variables usually entering into the uncertainties of engine designing. This is especially desirable when engines differing materially in the percentages of clearance are being compared

In the accompanying table values of the ratios of expansion and of a factor for the rapid determination of the true mean

effective pressure, according to equation (1), are given for varying values of clearance and points of cutoff.

With the foregoing principles established, we may proceed to the method of designing compound engines, and for the sake of greater clearness numerical quantities will be used.

Let it be required to design a compound engine condensing, to develop 500 indicated horsepower, with steam at 140 pounds per square inch absolute pressure at the high-pressure steam chest, and with a mean piston speed of 600 feet per minute. The total number of expansions will be taken as 16*, giving a theoretical terminal pressure in the low-pressure cylinder of $140 \div 16 = 8.75$ pounds per square inch, absolute. Back pressure in the low-pressure cylinder will be taken as 3 pounds per square inch, the ratio of cylinder volumes 1 to 4, and the clearance 5 percent of the piston displacement in each cylinder. Drop from the high-pressure cylinder into the receiver will be taken as 8 pounds, and a further drop of pressure before the steam enters the low-pressure cylinder of 4 pounds will be allowed for. This determines the initial steam pressure for the low-pressure cylinder.

The theoretical indicator card may first be drawn as in Fig. 2, on which the values are designated as follows:

V_1 = volume of piston displacement in the high-pressure cylinder.

V_2 = volume of piston displacement in the low-pressure cylinder.

v_1 = volume of clearance in the high-pressure cylinder.

v_2 = volume of clearance in the low-pressure cylinder.

X_1 = percent of stroke traversed by the piston at cutoff in high-pressure cylinder.

X_2 = percent of stroke traversed by the piston at cutoff in low-pressure cylinder.

The total volume of the low pressure cylinder, including clearance, must be 16 times the volume of steam in the high-pressure cylinder at the point of cutoff, which initial volume must include the high-pressure clearance. These two volumes may thus be represented by 16 and 1, respectively. Also, since a cylinder ratio of 1 to 4 has been assumed, with the percentage of clearance the same in both cylinders, the total volume of the high-pressure cylinder including clearance will be one-fourth of 16, or 4. The number of expansions in the high-pressure cylinder will thus be 4.

We now have:

$$V_2 + v_2 = V_2 + .05V_2 = 1.05\ V_2 = 16$$

$$\text{Hence} \qquad V_2 = \frac{16}{1.05} = 15.24$$

$$v_2 = .05 \times 15.24 = 0.76$$

$$V_1 = \frac{15.24}{4} = 3.81$$

$$v_1 = .05 \times 3.81 = 0.19$$

$$X_1 = \frac{1 - v_1}{V_1} = \frac{1 - .19}{3.81} = 0.21 \text{ nearly,}$$

which gives the point of cutoff in the high-pressure cylinder.

P_2 = terminal pressure in high-pressure cylinder $= \frac{140}{4} = 35$ pounds per square inch.

P_{b1} = back pressure in high-pressure cylinder $= 35 - 8 = 27$ pounds per square inch.

P_3 = initial pressure in low pressure cylinder $= 27 - 4 = 23$ pounds per square inch.

Since the steam which fills the high-pressure cylinder and its clearance space at 35 pounds pressure is to be contained in the low-pressure cylinder and clearance space at the point of cutoff at 23 pounds pressure, it is necessary to know the volume which the steam will occupy at the two pressures. These volumes will be inversely proportional to the densities at the two pressures, or directly as the specific volumes. From the steam tables, one pound of steam at 35 pounds pressure occupies 11.78 cubic feet, and at 23 pounds, 17.46 cubic feet.

$$\text{Therefore,} \quad \frac{X_2}{100} \times (V_2 + v_2) = \frac{17.46}{11.78}(V_1 + v_1)$$

$$\text{or,} \quad \frac{X_2}{100} \times (15.24 + .76) = \frac{17.46}{11.78} \times 4$$

whence, $X_2 = 33.9$ percent, nearly, which indicates the point of cutoff in the low-pressure cylinder.

The ratio of expansion in the low-pressure cylinder may now be found to be,

$$r_2 = \frac{V_2 + v_2}{\frac{X_2}{100}(V_2 + v_2)} = \frac{16}{.339 \times (15.24 + .76)} = 2.7.$$

Or, by ratio of pressures, $r_2 = \frac{P_3}{P_4} = \frac{23}{8.75} = 2.7$, nearly.

The theoretical mean effective pressures for both cylinders may now be calculated in the regular manner, from equation (1).

m. e. p. in high-pressure cylinder

$$= \frac{P_1 \times X_1 \times V_1 + P_1 (X_1 V_1 + v_1) \text{ hyp. log. } 4}{V_1} - P_b$$

$$= 140 \left(\frac{.21 \times 3.81 + (.21 \times 3.81 + .19) \log. 4}{3.81}\right) - 27$$

$= 53.36$ pounds per square inch.

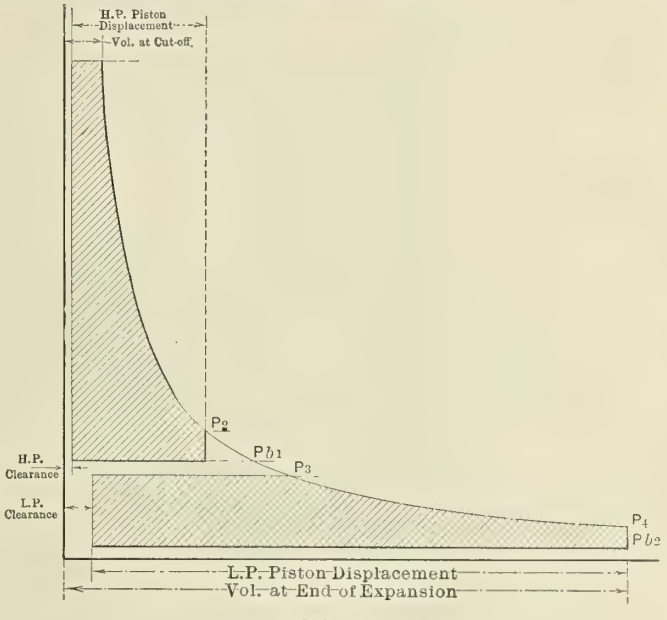


FIG. 2.

m. e. p. in low-pressure cylinder

$$= 23 \left(\frac{.339 \times 15.24 + (.339 \times 15.24 + .76) \log. 2.7}{15.24}\right) - 3$$

$= 13.68$ pounds per square inch.

The actual mean effective pressures that may be expected in the two cylinders are considerably smaller than these values. The factor to be used, which corresponds to the factor commonly employed in usual methods of design, and known as the card factor, must, of course, be decided upon after due consideration of all the conditions; but it is not in the scope of the present article to discuss that point. For the present instance it will be assumed that the same factor applies to both cylinders, and 80 percent will be used. Therefore we have for the expected mean effective pressures,

in the high-pressure cylinder, m. e. p. $= .80 \times 53.36 = 42.688$ pounds per square inch.

in the low-pressure cylinder, m. e. p. $= 0.80 \times 13.68 = 10.944$ pounds per square inch.

The m. e. p. of the low-pressure cylinder, referred to the high-pressure cylinder, will be $10.944 \times 4 = 43.776$ pounds per square inch.

*This is much higher than is commonly met in marine practice.—Ed.

Hence the total power of the engine will be divided between the cylinders in the ratio of 42.688 for high-pressure cylinder to 43.776 for low-pressure cylinder. Hence the power of the high-pressure cylinder is,

$$500 \frac{42.688}{(42.688 + 43.776)} = 247, \text{ and for}$$

the low-pressure cylinder $= 500 - 247 = 253$.

The power to be developed in each cylinder, the mean effective pressure, and the piston speed being thus determined, the diameters of the two cylinders may be calculated in the usual manner, with the equation,

$$\text{I. h. p.} = \frac{PLAN}{33{,}000}$$

wherein $L \times N =$ length of stroke times number of strokes per minute, is the mean piston speed. We have, therefore, for high-pressure cylinder,—

$$247 = \frac{42.688 \times 600 \times A}{33{,}000}$$

from which, area of high-pressure piston $= A = 318.24$ square inches.

In a similar manner we have for the low-pressure cylinder,

$$\text{Area of piston} = A_2 = \frac{33{,}000 \times 253}{10.944 \times 600}$$

$$= 1271.5 \text{ square inches.}$$

Or, since the ratio of cylinders has been taken 4 to 1,

Area of low-pressure piston $= 318.24 \times 4 = 1272.96$ square inches.

The diameters giving the areas nearest to these values may be found in any convenient table of areas of circles, and the length of stroke and number of revolutions fixed upon according to conditions in such proportions as to give the piston speeds called for. Diameters are 20⅛ and 40¼ inches, neglecting influence of size of piston rods on crank or lower end.

It is believed that the above method, as outlined, has certain advantages over methods usually given, although strict accuracy with any method can not be expected. With saturated steam the point of cutoff in the low-pressure cylinder would, in practice, be varied somewhat in order to hold the receiver pressure at the desired point, but with the receiver jacketed the value fixed by the theoretical discussion would probably be nearly correct.

Independent determination of the card factor to give the mean effective pressure that may confidently be expected in the actual engine would be necessary, in order to use the method to best advantage, and this should be done for the high and low-pressure cylinders separately. With these values carefully selected good results may be expected.

The chief advantage of the system lies in the fact that the two cylinders are considered separately. This enables the designer to provide a steam distribution that will be best suited to the conditions of operation in each. Since the two cylinders are really independent engines connected to the same shaft, they are properly treated as such, and all of the features affecting economy of simple engines working with steam of the same pressures apply to them. Drop of pressure out of the high-pressure cylinder is thus desirable for the same reasons that make it inadvisable to run simple engines expanding to the back-pressure line, and it is desirable to treat this feature in a rational manner. Evidently by a continuation of the process triple and quadruple engines may be designed by the same method.

Ratios of expansion and factors for theoretical mean effective pressures in steam engines for varying points of cut-off and clearance volumes. Determined as follows :—

Let P_1 = initial pressure of steam.
P_b = back pressure on piston.
L = length of stroke.
l = distance traversed by piston at cut-off.
c = length of cylinder to equal clearance volume.

Then ratio of expansion $= r = \frac{L + c}{l + c}$

$$\text{m. e. p.} = P_1 \left(\frac{l + (l + c) \text{ hyp. log. } r}{L}\right) - P_b$$

Let $\left(\frac{l + (l + c) \text{ hyp. log. } r}{L}\right) = F$

Then theoretically m. e. p. $= (F \times P_1) - P_b$

VALUES OF F AND r.

Cut off	Clearance in percent of piston displacement.																			
	2%		2.5%		3%		4%		%		6%		7%		8%		9%		10%	
	r	F	r	F	r	F	r	F	r	F	r	F	r	F	r	F	r	F	r	F
1/8	7.034	.408	6.833	.413	6.646	.419	6.303	.429	6.000	439	5.730	.448	5.487	.457	5.269	.466	5.070	.474	4.889	.482
1/7	6.263	.442	6.197	.446	5.956	.451	5.638	.471	5.444	470	5.225	.478	5.027	.487	4.847	.495	4.681	.502	4.530	.510
1/6	5.464	.484	5.348	.488	5.237	.492	5.033	.501	4.848	.509	4.676	.516	4.521	.524	4.379	.531	4.247	.538	4.125	.545
1/5	4.687	.537	4.555	.541	4.478	.545	4.333	.552	4.200	.559	4.080	.566	3.963	.572	3.858	.578	3.759	.584	3.667	.590
1/4	3.778	.609	3.727	.612	3.680	.615	3.586	.620	3.500	.626	3.420	.631	3.344	.636	3.273	.641	3.216	.646	3.143	.651
1/3	2.887	.707	2.860	.710	2.835	.712	2.786	.716	2.741	.720	2.695	.723	2.653	.727	2.613	.731	2.575	.734	2.540	.737
2/5	2.430	.773	2.410	.774	2.396	.776	2.364	.779	2.333	.781	2.304	.784	2.277	.789	2.280	.790	2.225	.792	2.200	.794
1/2	1.962	.850	1.952	.851	1.942	.852	1.926	.854	1.910	.856	1.892	.857	1.877	.859	1.860	.860	1.847	.862	1.833	.863
2/3	1.487	.938	1.482	.939	1.479	.939	1.472	.940	1.465	.940	1.459	.941	1.453	.942	1.446	.942	1.440	.943	1.435	.944
3/4	1.325	.966	1.323	.967	1.320	.967	1.316	.967	1.313	.968	1.309	.968	1.305	.968	1.301	.968	1.298	.969	1.296	.970
4/5	1.244	.979	1.242	.979	1.241	.979	1.236	.979	1.335	.979	1.23	.980	1.230	.980	1.227	.980	1.225	.981	1.222	.981

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports under date of October 10, 1906, the following percentage of completion of vessels building for the United States Navy:

			Sept. 1	Oct. 1
BATTLESHIPS.				
Nebraska	19 knots.	Moran Brothers Co.	96.38	96.71
Georgia	19 "	Bath Iron Works	99.6	100.
Connecticut	18 "	Navy Yard, New York, N. Y.	99.08	99.7
Vermont	18 "	Fore River Shipbuilding Co.	92.7	93.6
Kansas	18 "	New York Shipbuilding Co.	87.	91.
Minnesota	18 "	Newport News Co.	95.	96.
Mississippi	17 "	Wm. Cramp & Sons	62.18	65.52
Idaho	17 "	Wm. Cramp & Sons	60.29	62.41
New Hampshire	18 "	New York Shipbuilding Co.	51.8	53.3
ARMORED CRUISERS.				
California	22 knots.	Union Iron Works	95.	95.7
South Dakota	22 "	Union Iron Works	93.	93.2
North Carolina	22 "	Newport News Co.	57.18	61.09
Montana	22 "	Newport News Co.	51.03	54.28
SEMI-ARMORED CRUISERS.				
Milwaukee	22 knots.	Union Iron Works	98.5	98.9
SCOUT CRUISERS.				
Chester	24 knots.	Bath Iron Works	44.97	47.4
Birmingham	24 "	Fore River Shipbuilding Co.	44.9	47.4
Salem	24 "	Fore River Shipbuilding Co.	45.	47.5
SUBMARINE TORPEDO BOATS.				
Cuttlefish	.....	Fore River Shipbuilding Co.	84.9	89.
Viper	.....	Fore River Shipbuilding Co.	73.	73.
Tarantula	.....	Fore River Shipbuilding Co.	85.9	90.
Octopus	.....	Fore River Shipbuilding Co.	69.9	74.4

THE TURBINE-PROPELLED STEAMER KAISER.

BY DR. ALFRED GRADENWITZ.

The Hamburg-American Line has been the first among German steamship companies to adopt turbine propulsion for one of their express steamers, the *Kaiser,* built at the Vulkan Shipyards at Stettin, being intended for regular service between Hamburg and the German North Sea bathing-places. This steamer was completed in the short interval of fourteen months. She is 96 meters (315 feet) in total length, 11.65 meters (37.2 feet) in breadth, and with a draught of 3.03 meters (10.8 feet), has a displacement of 1,923 tons. She is able to accommodate 2,000 passengers. Four water-tube boilers constructed by the Vulkan firm have been provided, supplying steam at 15 atmospheres pressure (220 pounds per square inch). The stoking is operated on the Howden forced draught system, the air for combustion being preheated in a regenerator by the escaping furnace gases, and the feed water by the exhaust steam of the auxiliary engines.

STERN VIEW OF TURBINE-PROPELLED EXPRESS STEAMER KAISER.

The turbine plant, constructed by the Allgemeine Elektrizitäts Gesellschaft, comprises two units of an aggregate of 6,000 horsepower, each of which operates a shaft with one propeller. The turbines are pressure stage turbines on the Curtis system, the rotary part comprising five multiple-rim paddle wheels, with an equal number of pressure stages, to which a drum with a large number of paddle rims is connected. After at first working in the five paddle wheels, the steam acts on the rims of the low-pressure drum, leaving the last rim to enter the condenser through the exhaust compartment.

A backward turbine working in two pressure stages (each of which comprises several rims) has been arranged on the same shaft in the same casing.

After having been designed in July, 1904, the first of these turbines was, at the end of March, 1905, taken to the testing floor of the turbine works, where the two units with condensers and auxiliary engines were submitted to rather severe tests, at heavy overloads. Tests were made with superheated steam and with different vacua, using as load in the place of the propellers a specially constructed hydraulic brake.

A six hours' delivery test at full load by means of the water brake was finally carried out on each turbine at 600 revolutions per minute. A weight of 70 tons had been allowed for each unit; but the complete turbine, inclusive of the bearings, thrust bearings and oil pumps, in addition to all the pipes required for oil and draining, worked out at a weight of only 57 tons. After having been sent to Stettin as a whole, the turbines were lowered by means of cranes into the engine room of the steamer, and after having there been fitted, a short test was carried out with uncoupled propeller, showing that the engines worked quite smoothly on the ship-foundation.

The steamer first went to sea on August 16, 1905, running a number of trial trips in the Baltic Sea up to the 10th of September, which trips were intended to ascertain the most favorable propeller construction for the given output of 3,000 horsepower and with 600 revolutions per minute. In the extremely short interval of less than four weeks, trial trips were carried out with four different propellers, which was rendered possible

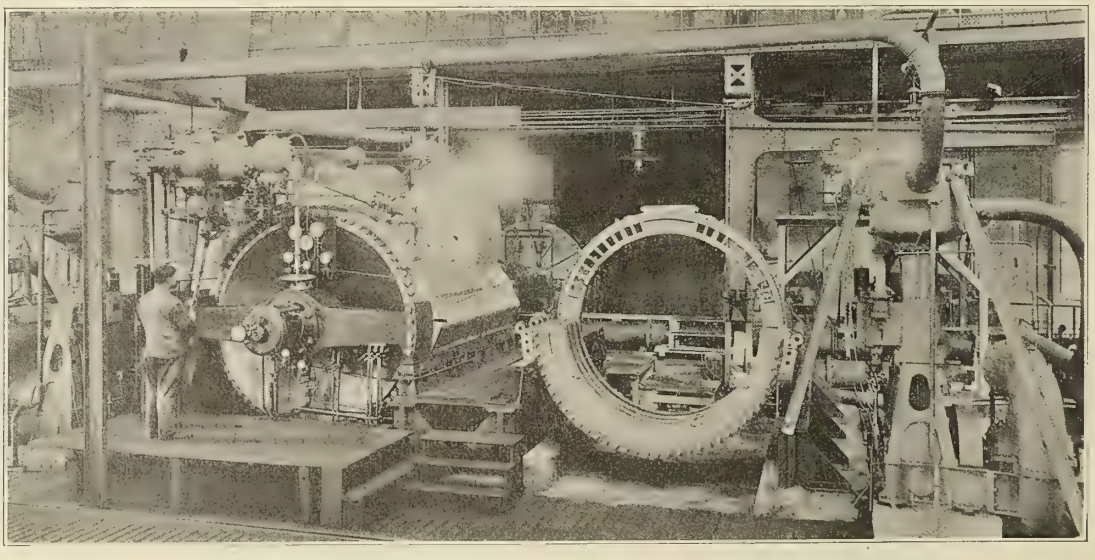
only by the fact that not the least hitch occurred in the turbine plant throughout these trips.

The final trial trip was carried out with the propellers finally chosen, in rather unfavorable weather, when the ship made, as an average of two hours, 20.46 knots, as against a guaranteed speed of 19.5 knots. The total consumption of coal had been guaranteed at 4.7 tons per hour with a speed of 19.5 knots, corresponding to figures which should have been obtained for triple expansion reciprocating engines. During the two hours' coal consumption test the total consumption of coal, including that for all the accessory machinery, with 20.05 knots, was found to be only 4.06 tons per hour, working out at 0.66 kilograms (1.46 pounds) per horsepower hour. Now this economy is 20 percent greater than that guaranteed. It should also be mentioned that the operation of the engines has been considerably simplified, only the auxiliary machinery requiring any attendance worth speaking of.

The simplicity and ease of control of the turbines is especially remarkable, and has been acknowledged by all experts having had an opportunity to inspect them; in fact any reversing of steam paths for different speeds has been avoided, only a single controlling valve being provided for regulation from zero to the full number of revolutions, the same valve operating the astern turbine after reversing a slide. The efficiency of the astern turbine is so satisfactory that while being very easily controlled, the steam consumption is only slightly higher than that of the forward turbine, thus maintaining the full boiler pressure even in the case of a continued backward course. The time required from "full speed ahead" to the backward course is about the same as in the case of equivalent reciprocating engine vessels.

THE HAMBURG-AMERICAN TURBINE STEAMER KAISER.

Turbo-dynamos of the A. E. G.-Curtis system are used for lighting the steamer, all the lamps on deck, in the saloons, dining-halls, cabins, etc., deriving their current from two units of 20 kilowatts each. The same machines yield the current for the operation of the smaller auxiliary machinery, such as laundry machines, refrigerating machinery, etc.



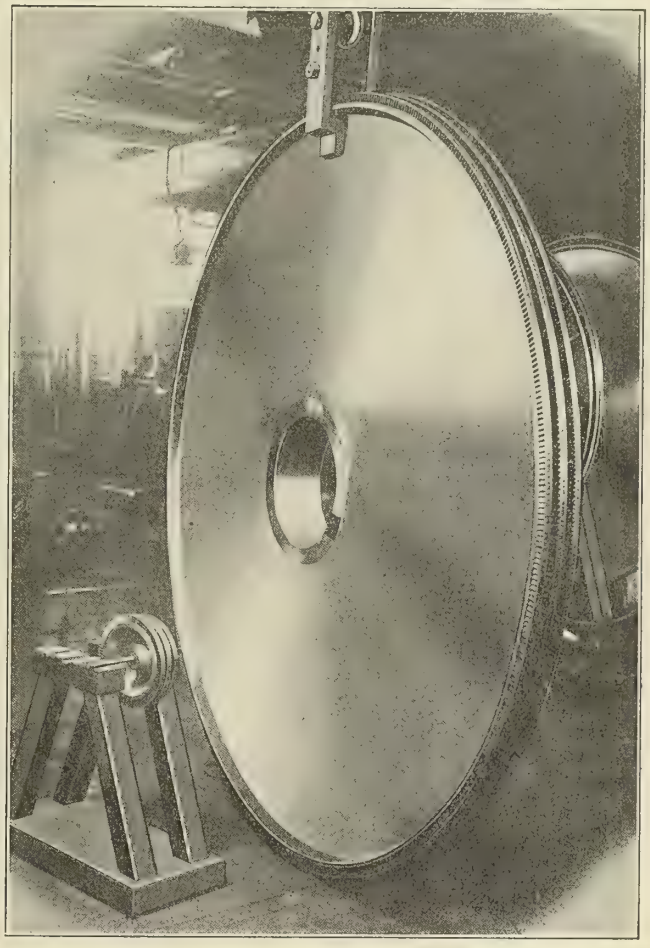
ASSEMBLED 3,000-HORSEPOWER TURBINE ON TESTING FLOOR. ANOTHER BEING ASSEMBLED.

The turbines, direct coupled to the dynamo, are single-stage turbines with three speed stages, thus possessing only a single wheel provided with three rims of vanes. The dynamo has been arranged between two bearings, outside of which the casing of the turbine has been fitted. The turbine wheel is made from best class steel, and has been fitted to the end of the dynamo shaft, thus insuring a high simplicity and stability, in addition

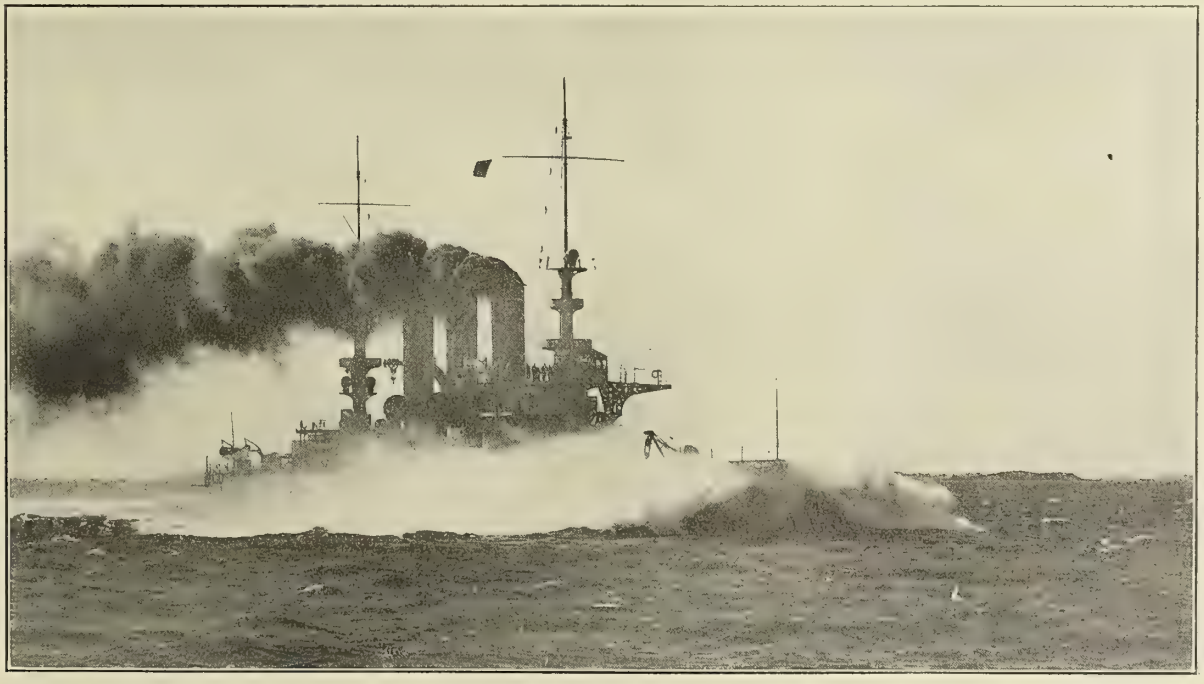
to a small weight. After entering through the main throttle valve into the steam-regulating compartments, the steam gets to the nozzles, in which the pressure is entirely converted into speed. The spring regulator fitted to the free shaft end acts through a rod on a throttling valve in the steam compartment. The turbines are connected to the central condensing plant. These small machines are operated with exhaust during the time the steamers are in the harbor, thus dispensing with the operation of the large central condensing plant. The nozzles especially provided for exhaust are closed by hand in case the turbines are to be operated for condensing in continual service. The bearings receive their oil from a valveless wheel pump under pressure, the bushes being cooled with water.

The dynamo has been designed as shunt dynamo, possessing two auxiliary poles, to avoid any sparking of the brushes in the case of variable loads. It is able to run from zero to full load without any displacement of the brushes, which are of carbon.

After having been taken over by the Hamburg-American Line on September 23, 1905, the *Kaiser* performed some trips to Helgoland and Sylt in the course of the ensuing weeks. In connection with these trips, several of which were performed in very bad weather, the steamer proved superior to the ships propelled by reciprocating engines. With a heavy north-west gale (wind intensity 10) the *Kaiser* was able to make headway against the wind at a speed of 17 knots, being the last ship that could enter the harbor against the storm. This satisfactory behavior should be ascribed both to the shape and construction of the hull, and to the use of the turbines for the propulsion of the vessel. Whenever, owing to the pitching of the ship, the screws issued from the water, which phenomenon is very frequent in the case of stormy weather, the behavior of the vessel was especially remarkable. In fact, whereas in the case of reciprocating engines a large increase in the number of revolutions is observed, resulting in heavy shocks to the hull (to offset which effect the steam should be throttled in such cases, thus requiring a permanent attendance) only a very slight increase in the number of turns was observed with the turbines of the *Kaiser,* necessitating no steam throttling. The smooth running of the vessel was not interfered with in any way, nor was the issuing of the screws at all noted by the passengers.



LOW-PRESSURE END OF ROTOR OF PROPELLING TURBINE AND DYNAMO ROTOR.



THE BATTLESHIP VIRGINIA THROWING UP A HUGE MASS OF SPRAY AND FOAM.
(Copyright, 1906, N. L. Stebbins.)

DEVELOPMENT OF THE MARINE STEAM TURBINE.*

BY HON. C. A. PARSONS AND R. J. WALKER.

During the last few years the steam turbine has formed the subject of many papers read before various leading institutions, and its different applications have often been referred to. It is hardly within the scope of this paper to deal in detail with the earlier experiments and trials which have led up to the present satisfactory position of the turbine. It may be of interest, however, to briefly describe the various steps that have been made from time to time in the development.

In 1884 the first Parsons compound steam turbine engine was built, and was applied to the driving of a dynamo. This engine was designed for 10 horsepower, with a modified high-speed dynamo, and for a working speed of 18,000 revolutions per minute. It ran for some years, doing useful work, and is now in South Kensington Museum. Subsequently, efforts were made towards the construction of engines of larger size, which resulted, in the year 1888, in several turbo-alternators of 120 horsepower being supplied for the generation of current in electric light stations, and at that period the total horsepower of turbines at work reached in the aggregate about 4,000, all of which were of the parallel flow type and non-condensing.

In 1892 the steam turbine was first adapted to work in conjunction with a condenser. This engine was capable of developing 200 horsepower at a speed of revolutions of 4,800 per minute, and drove an alternator of 150-kilowatt output. It was tested by Professor Ewing, F. R. S., and the result of the test showed a consumption of steam of 27 pounds per kilowatt hour, which is equivalent to about 16 pounds per I. H. P., with steam moderately superheated, and a vacuum of 28 inches. Since then, various improvements have been made, until at the present time the steam turbine is generally recognized to be an efficient and practical engine, which, in the larger sizes, has attained a high degree of economy in steam.

The following table shows the advance made in sizes and increased economy in steam consumption as applied for electrical purposes:

Date.	Capacity kilowatts.	Steam per kilowatt hour, lbs.	Vacuum inches.	Superheat degrees F.	Steam pressure per square inch, lbs.
1887	75	50.	....	...	120
1892	100	27.	27	50	...
1898	1250	18.81	28	180	130
1901	1000	17.30	27	198	150
1902	3000	14.74	27	235	138
1904	4000	15.40	28.7	150	200

Superheating of the steam has been adopted in the case of land turbines, and it has been found by experiments that for every 10 degrees F. of superheat, the steam consumption is reduced by about one percent.

Fig. 1, which is based upon a large number of tests, shows the steam consumption of different sized plants per kilowatt-hour without superheating, from which it will be noted that the efficiency increases steadily as the size grows larger.

Turbine engines are also used for generating electrical current for the transmission of power, the working of electric tramways, railways, electric pumping, coaling and similar purposes. They are also used for coupling direct to fans, and driving them for producing forced and induced draft for general ventilating purposes; also for driving centrifugal and screw pumps, and also for the more important work of driving blowers for supplying air under pressure, for blast furnaces and other purposes.

Land turbines have been very largely adopted in the past few years for the generating of electricity, judging from the fact that these machines are running in England in the power stations of some forty corporations and of twenty-five public electric supply companies. One of the largest power stations in England, now approaching completion, contains two units of 3,000 horsepower and six units of 6,000 horsepower each. The total horsepower of turbines of the Parsons type delivered and on order, including turbines at work and under construction by licenses on the continent and in the United States at the present time, for land purposes is nearly 2,000,000. The installations now on order include units of upwards of 10,000 horsepower.

The application of the steam turbine to the propulsion of ships has attracted a great deal of attention in the shipbuilding world within the last few years.

It was not until the year 1894 that the idea of propelling a vessel by means of a steam turbine was put into practical form. The *Turbinia,* as is now generally known, was the first vessel to be fitted with turbine engines, and between the years 1894 and 1898 many experiments were made with the *Turbinia,* necessitating radical changes in the design and arrangement of the machinery. The first engine which was tried was of the radial flow type, giving about 1,500 horsepower to a single screw. The results, however, were far from satisfactory, a speed of only 18 knots being obtained. Several different propellers were tested with this engine, and the results compared with the power registered by a dynamometer showed in every case a very low propeller efficiency. The original turbine engine was removed, and the engines finally adopted consisted of three turbines in series; high-pressure, intermediate-pressure and low-pressure, each driving a separate shaft, with three propellers on each shaft.

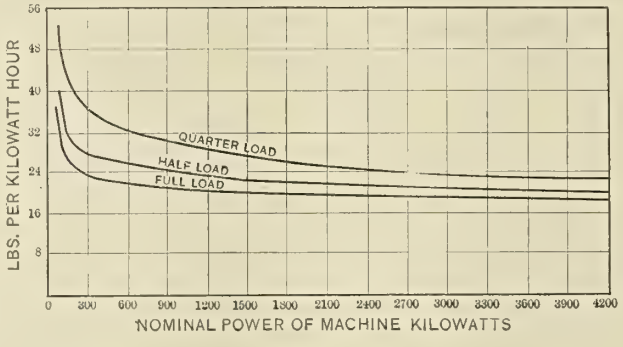


FIG. 1.

A reversing turbine was coupled with the low-pressure turbine to the central shaft. Very exhaustive trials were carried out by Professor Ewing in 1897. A full account of these trials will be found in the *Transactions of the Institution of Naval Architects,* Vol. XLV (1903).

Following the success of the *Turbinia,* the torpedo boat destroyers *Viper* and *Cobra* were built and fitted with turbine machinery for the Royal Navy, and achieved remarkable speeds, the *Viper* taking the position of being the fastest vessel in the world, having attained the phenomenal speed of 36.88 knots. Unfortunately, however, the *Viper* ran on the rocks of the Channel Islands in a fog, and ultimately became a total wreck. The *Cobra* foundered in a storm. Thus, after two or three years of hard work, the *Turbinia* was the only vessel afloat fitted with turbine engines.

About this period very great difficulty was experienced in endeavoring to induce railway companies and owners of mercantile vessels to build a turbine boat. Each company appeared anxious that someone else should make the first experiment.

The marine turbine was first adopted for commercial purposes in the Clyde steamer *King Edward,* to the order of Captain Williamson, in the summer of 1901. So successful was this vessel during the first season's running on the Clyde (the year of the great Exhibition at Glasgow), that an order was placed for a second vessel, *Queen Alexandra,* and the performance of these two vessels running on the Firth of Clyde demonstrated the commercial advantages accruing from the adoption of the turbine system. Other vessels quickly followed the *King Edward* and *Queen Alexandra,* until there are at the present time thirty-one turbine vessels in service for commercial purposes, representing

*Paper read before the Engineering Congress at the Olympia Engineering Exhibition, London.

a total of about 105,000 gross tonnage and 235,000 indicated horsepower.

The following table will perhaps best represent the various steps in the adoption of the turbine engine for commercial purposes:

Name of vessel and owners.	Dimensions.	Gross tonnage.	Approx. I. H. P.	Year built.
King Edward—				
Turbine Steamers, Ltd.	250x30x17¾	562	3,500	1901
The Queen—				
S. E. & Chat. Ry. Co.	310x40x25	1,676	7,500	1903
Virginian—				
Allan Line	520x60x41	10,754	12,000	1904
Carmania—				
Cunard Line	678x72x52	19,524	21,000	1905
Lusitania—				
Cunard Line	785x88x60½	32,500	70,000	1906

As regards war vessels, the destroyers *Velox* and *Eden* were the next vessels following the *Viper* to be built for the British Admiralty. These two vessels were fitted with additional engines for obtaining economical results when cruising at low speeds. The next vessel to be fitted with turbines for the British Admiralty was the third-class cruiser *Amethyst*. This vessel was also fitted with cruising turbines, and the results of the trials, as compared with sister vessels ordered at the same time as the *Amethyst*, and of the same dimensions and lines, but fitted with reciprocating engines, demonstrated the economy of the Parsons marine turbine as fitted in this class of vessel, and more especially at the higher powers.

	Princesse Elisabeth.	*Princesse Clementine.*	*Marie Henriette.*	*Leopold II.*
Length between perpendic.	104m.85	103m.70	103m.70	103m.70
Breadth	12m.192	11m.58	11m.58	11m.58
Mean draft	2m.92	2m.85	2m.82	2m.82
Displacem't in metric tons..	2,005	1,853	1,847	1,829
Registered tons	1,747	1,474	1,450	1,375
Type of engines	Turbines	Compound	Compound	Compound
Speed on trial, knots	24	22.187	22.2	22
Date of construction	1905	1896	1893	1893

In the year 1905 the mean consumption of coal per single trip and the mean time per trip from Ostend to Dover and vice versa, of the four preceding vessels, were as follows:

	P. E.	P. C.	M. H.	L. II.
Total number of trips	82	278	278	232
Mean duration of trips, minutes	187	217	212	227
Mean consumption per trip, tons	23.01	24.05	23.82	24.30

For the six first months of the year 1906 the corresponding results were the following:

	P. E.	P. C.	M. H.	L. II.
Total number of trips	134	132	106	44
Mean duration of trips, minutes	185	210	206	202
Mean consumption per trip, tons	22.71	23.22	24.27	24.87

	Turbine boat.	Mean result paddle steamers.
Total number of trips	216	1,070
Mean duration of trip, minutes	186	215
Mean consumption per trip, tons	24.06	24

From the above table, it will be seen that the turbine boat does the passage in about 15 percent less time than the paddle steamers, on the same coal consumption. To reduce the turbine

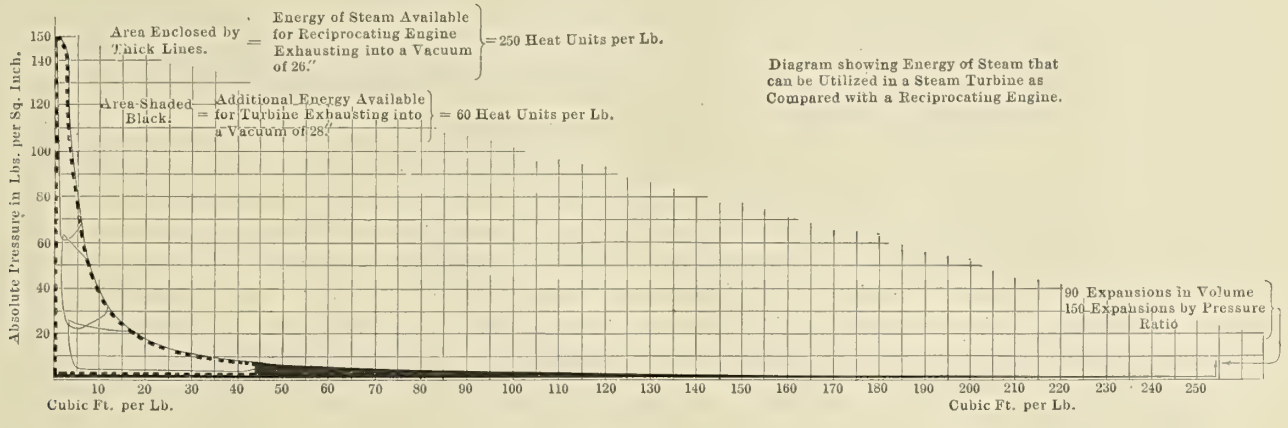

Area Enclosed by Thick Lines. = Energy of Steam Available for Reciprocating Engine Exhausting into a Vacuum of 26." = 250 Heat Units per Lb.
Area Shaded Black. = Additional Energy Available for Turbine Exhausting into a Vacuum of 28." = 60 Heat Units per Lb.
Diagram showing Energy of Steam that can be Utilized in a Steam Turbine as Compared with a Reciprocating Engine.
Absolute Pressure in Lbs. per Sq. Inch.
90 Expansions in Volume
150 Expansions by Pressure Ratio
Cubic Ft. per Lb.
Cubic Ft. per Lb.

FIG. 5.

The battleship *Dreadnought* is one of the latest war vessels to be fitted with turbine engines. This vessel's turbines, which were manufactured by the Parsons Marine Steam Turbine Company, Limited, at the Turbinia Works, have now been installed, and the vessel is being prepared for trials. Particulars are given in the Memorandum of the Navy Program of new construction, of the various war vessels which are at the present time being built, some of which are now nearing completion.

Comparisons of the earning powers of turbine vessels have been made from time to time with similar vessels on the same respective routes, which have been found to be favorable to the turbine,and in some cases the saving in coal is very considerable. A full account of these comparisons will be found in the *Transactions of the Institution of Civil Engineers*—"The Steam Turbine," by the Hon. C. A. Parsons and G. Steney,—and in the *Transactions of the Liverpool Engineering Society*—"Progress made in the Application of the Parsons Turbine to Marine Propulsion," by R. J. Walker.

By the courtesy of Monsieur Pierrard of the Belgian Government, particulars are just to hand of the first season's running of the turbine steamer *Princesse Elisabeth* on the Ostend and Dover service, and are given in the following table, along with the particulars of similar vessels on the same service, but fitted with paddle engines:

boat to the displacement and speed of the paddle boats, and assuming that the indicated horsepower varies as the cube of the speed, the mean consumption of the *Princesse Elisabeth* would be about 17 tons, as against 24 tons in the paddle boats, thereby showing a saving of over 25 percent.

The following mercantile vessels have been completed and placed on service this year:

The pleasure and mail steamer *Rewa* for the British India Steam Navigation Company, Ltd., of 455 by 56 feet and 16½ knots speed; the Clyde passenger steamer *Duchess of Argyll*, for the Caledonian Steam Packet Company, of 250 by 30 feet and 20 knots speed; the Thames passenger steamer *Kingfisher* for the General Steam Navigation Company, of 275 by 32 feet and 20 knots speed. The three cross-channel steamers for the Great Western Railway Company's new route, Fishguard to Rosslare, viz., *St. David, St. Patrick* and *St. George*, of 350 by 41 by 13½ feet and 22½ knots speed; the cross-channel steamer *Viper* for Messrs. G. & J. Burns, of 315 by 39½ by 12 feet and 21¾ knots speed; and another Clyde passenger steamer, the *Atalanta*, for the Glasgow and South Western Railway Company, of 210 by 27 by 10½ feet, and 17½ knots speed. In addition to the above, the two Great Central Railway steamers are nearing completion, and will, it is anticipated, be put on service this year, and the Khedival yacht *Mahroussa*, which has been

re-engined by Messrs. A. & J. Inglis, has run her trials, and a speed of 17½ knots was attained.

In addition to the large number of war vessels that are now being built and fitted with turbine machinery by some eight of the Parsons Marine Company's licenses, and also in addition to the two express Cunarders by Messrs. J. Brown & Company and Messrs. Swan, Hunter & Wigham Richardson, Ltd., and the Wallsend Slipway & Engineering Company, the following mercantile orders are in hand: Two large ocean-going liners by the Fairfield Shipbuilding & Engineering Company; two cross-channel steamers for the South Eastern & Chatham Railway, similar to the *Onward* and *Invicta;* and a further steamer for the Union Steamship Company of New Zealand, by Messrs. Denny; two large sets of turbine engines for shipment to Japan by the Turbinia Company at Wallsend, and the Royal yacht, building by Messrs. Inglis, the turbine engines for which are being manufactured at the Turbinia Works, Wallsend, representing a total horsepower of work in hand of about 500,000.

Although the turbine itself has been described in various publications which are now being issued, it may be of interest to some of the members to briefly describe the principle of the Parsons turbine, and to mention a few points incidental to the application of the turbine system to different classes of vessels.

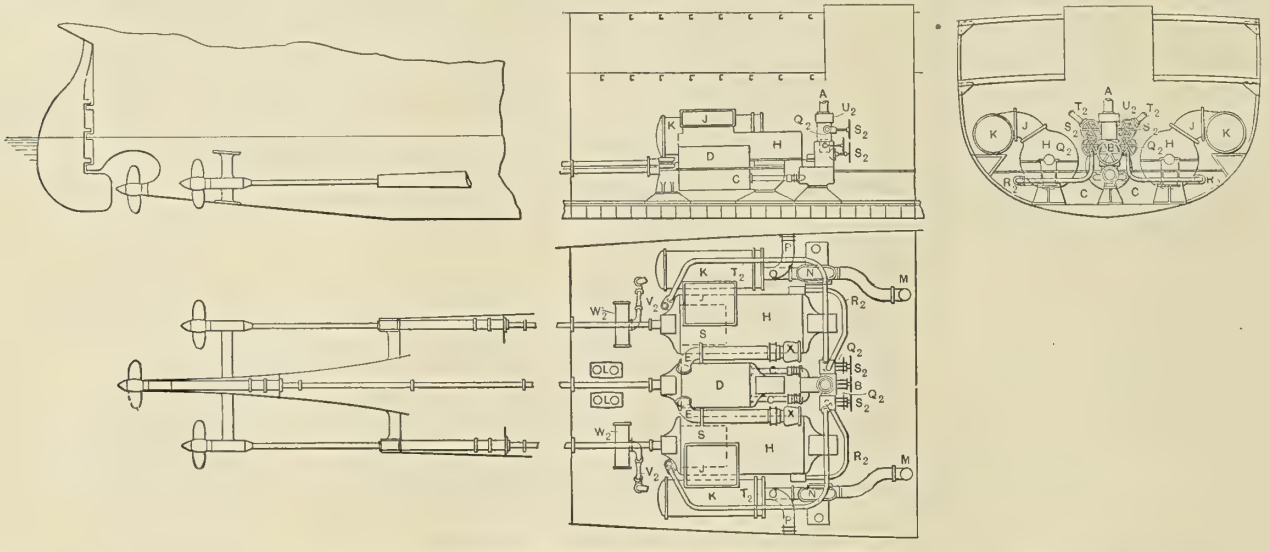


FIG. 2.—THREE-SHAFT ARRANGEMENT OF TURBINE MACHINERY.

A Steam from Boilers.
B Main Regulator Valve.
C Steam to High Pressure Turbine.
D High Pressure Turbine.
E Exhaust from H.P. Turbine to Automatic Closing Valve.
E_3 Exhaust from H. P. Turbine to I. P. Turbine.
E_4 Exhaust from H. P. Turbine to L. P. Turbine.
F Intermediate Pressure Turbine.
G Exhaust from I. P. Turbine to L. P. Turbine.
H Low Pressure Turbine.
J Exhaust to Condenser.
K Condenser.
L Air Pump.
M Main Inlet to Circulating Pump.
M_4 Steam to H. P. Astern Turbine.
N Circulating Pump.
N_4 High Pressure Astern Turbine.
O Circulating Discharge to Condenser.
O_4 Exhaust from H. P. Astern Turbine to L. P. Astern Turbine.
P Condenser Discharge Overboard.
P_4 Low Pressure Astern Turbine.
Q Astern Regulator Valve.
Q_2 Maneuvring Valve, L. P. Ahead.
Q_4 Regulator Valve to H. P. Cruising Turbine.
R Steam to Astern Turbine
R_2 Steam to L. P. Ahead Turbine.
R_4 Steam to H. P. Cruising Turbine.
S Astern Turbine.
S_2 Maneuvring Valve, L. P. Astern.
S_4 High Pressure Cruising Turbine.
T Cruising Regulator Valve.
T_2 Steam to Low Pressure Astern.
T_4 Exhaust from H. P. Cruising Turbine to Self-Closing Valve.
U Steam to Cruising Turbine.
U_2 Governor Valve.
V Cruising Turbine.
V_2 Vacuum Augmentor.
V_4 I. P. Cruising Turbine.
W Exhaust from Cruising Turbine to Automatic Closing Valve.
W_2 Vacuum Augmentor Condenser.
X Automatic Closing Valve.
Y Direct Steam Regulator Valve to I.P. Cruising Turbine.
Z Direct Steam to I.P. Cruising Turbine
a Exhaust from H. P. Cruising Turbine to Self-Closing Valve, when Engine Rooms are Independent.
b Self-Closing Valve.

References are to Figs. 2, 3 and 4.

In regard to the arrangement of turbines, this depends partly on the condition of service, and the various classes of vessels require, more or less, to be taken upon their own merits. The arrangement which has been almost universally adopted in the mercantile marine is that of dividing the power equally over three shafts, viz., a high-pressure on the center shaft, exhausting into two low-pressure turbines, one on each side of the vessel. Fig. 2 shows a three-shaft arrangement of machinery. In such a case the high-pressure turbine is made suitable for an expansion ratio of about five, and the low-pressure for twenty-five. The turbines are of the parallel flow type, the general course of the steam through them being parallel to the axis.

Each turbine consists of a number of rings of blades or vanes mounted on a drum or rotor. This rotor is inclosed within a cylindrical case, upon which rings of blades are also fitted. The rings of blades are alternately fixed and moving, that is to say, the blades in the cylinder are stationary or guide blades, while the blades fitted on the drum are moving blades. The diameter of the drum is less than that of the cylindrical case, and thus an annulus is left between the two, which is occupied by the blades. Steam issuing from the boilers flows through the first row of fixed or guide blades on the cylinder; it then, in jets of moderate velocity, impinges on the moving blades, and imparts to them a rotary motion, this process being repeated on each successive ring of fixed and moving blades throughout the turbine.

As the steam passes from ring to ring it falls in pressure and increases in volume, and to meet this expansion the blades are increased in height by steps. The areas of the steam passage through the blades thus gradually increase from one end of the turbine to the other, to correspond to the expansion in volume of the steam for the range of pressures from beginning to the end of the turbine.

The turbine shaft is coupled to the propeller shafting, and the thrust of the propeller is steam borne by the following arrangement:

Forward of the steam inlet a dummy piston is fitted, of reduced diameter to that of the main drum. A large number of small grooves are turned in this dummy piston, into which fit corresponding fixed brass rings, which are bedded into the dummy casing, forming steam baffles. The steam presses aft on the blades and on the annular part of the drum forming the step between the drum and the dummy piston, and the sum of these pressures balances the thrust of the propeller. To meet any unbalanced thrust, such as is set up when steam is turned on or off suddenly, and to maintain the true longitudinal alinement of the rotors, a small thrust block is provided at the forward end of the bearing. Where the turbine shaft passes through the casing, steam-packed glands are fitted. These glands consist of a number of rings or strips arranged in series, and designed to obtain a gradual rise or fall in pressure from the inner or steam end to the outer or atmospheric end of the gland. Valves are fitted to these glands to regulate the pressures, and to insure

against leakage of air inwards. All the main bearings are under oil pressure, and the oil is discharged from the pump at a pressure of from 8 to 10 pounds, or at such a pressure as to insure the efficient lubrication of all bearings.

In the exhaust casing of each of the low-pressure turbines a reversing turbine is fitted.

By means of suitable valves when maneuvering, the low-pressure and astern turbines on each side of the vessel are capable of being worked ahead or astern as required, independently of the high-pressure turbine, the latter, under such conditions, running idly in a vacuum. By this arrangement, the vessel has all the maneuvering qualities of a twin-screw vessel.

In torpedo boats, a similar arrangement is adopted as in the *Turbinia,* viz., three in series, Fig. 3, with one astern on the center shaft only. This arrangement permits of a reduced weight being obtained as compared with the high-pressure and two low-pressures in parallel for equal efficiency.

In very large powers, it is desirable to divide the power over four shafts (Fig. 4), thereby increasing the revolutions and reducing the size of the units. The four-shaft arrangement lends

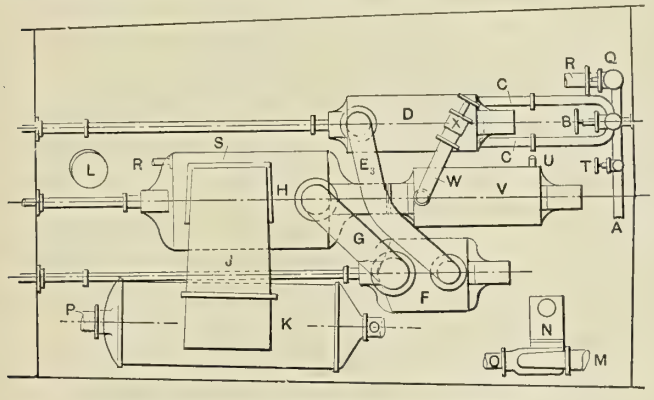


FIG 3.—ANOTHER THREE-SHAFT ARRANGEMENT.

itself well to large war vessels, as two complete sets of engines are obtained, one high-pressure and one low-pressure with condensing plant in each engine room. A two-shaft arrangement was adopted in the yacht *Narcissus,* which gave very good results; but for powers over 2,500 to 3,000 a two-shaft arrangement entails additional weight.

One of the chief difficulties which had to be contended with in applying the steam turbine to the propulsion of ships arose in connection with the propellers. It is desirable, for obvious reasons, that a turbine for a given efficiency should be designed to run at as high a rate of revolution as possible, consistent with propeller efficiency. The speed of turbines for land purposes is considerably higher than is permissible for marine purposes, where the speed of rotation is limited by considerations of propeller efficiency. The question of design of propeller, and turbine dimensions, require to be considered independently, and also to arrive at the best compromise to meet the conditions required, *i. e.*—as to weight, space, efficiency, and conditions of service. With a view of increasing the revolutions, the diameters of the propellers and pitch ratios are less than is usual in the case of ordinary reciprocating engines, the smaller diameter of propeller necessitating higher ratios of blade surface to disk area than hitherto attempted with ordinary propellers.

In some of the earlier vessels multiple propellers were tried, but subsequent experiments showed that single propellers on each shaft were preferable. The loss of efficiency which has been observed in some of the vessels fitted with multiple propellers appears to have been due partly to interference from the forward screws, and partly to cavitation. Although more light might yet be thrown on the question of high speed propeller efficiency, a considerable amount of experience has now been obtained with turbine driven propellers, which enables a close estimate to be made as to the efficiency which might be expected in a given design where reliable data as to horsepower necessary for the proposal in question can be obtained.

On account of the greater range of expansion dealt with in the turbine as compared with the reciprocating engine, a good vacuum is much more essential in the former than in the latter, and because of the importance of a high vacuum with the steam turbine, careful consideration requires to be given to the condensing plant. With a view of maintaining a high vacuum, a new apparatus has been introduced, to assist the ordinary air pump and condenser, known as the Vacuum Augmentor. The augmentor consists of a small steam jet placed in a contracted portion of a pipe, led from the bottom of the condenser. The jet draws air from the condenser, and delivers it to the air pump through a small auxiliary cooler. By this means the air is reduced to a negligible quantity. The vacuum augmentor has now been fitted in several vessels with very good results. As an illustration of the importance of high vacuum, Fig. 5 shows the energy of steam that can be utilized in a steam turbine, as compared with a reciprocating engine.

Another point which has often been referred to in connection with turbine machinery is the question of boiler pressure. The effect of differences of boiler pressure between 150 and 200 pounds is relatively smaller with turbines than with reciprocating engines, and it is very questionable, in the majority of cases, whether the saving in coal by the adoption of high boiler pressures would justify the increase. More especially is this the case in moderate speed vessels, where, to obtain reasonable propeller efficiency, a certain diameter of propeller is necessary, and the revolutions to correspond are moderately low. It is necessary to obtain a certain surface speed of turbine, as well as a certain number of rows of turbine blades, to obtain reasonable economy. The diameter of the turbine, therefore, becomes

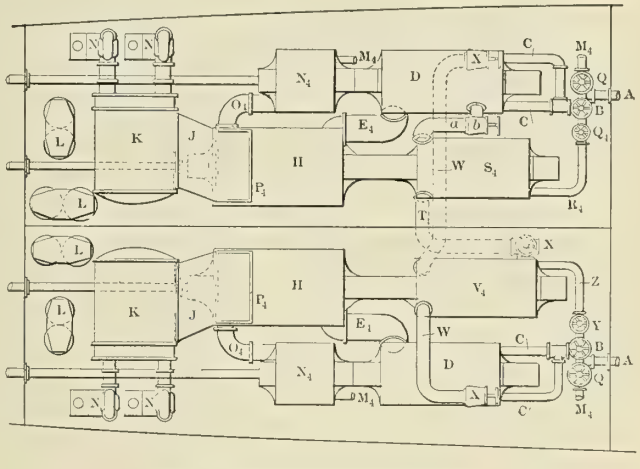


FIG. 4.—FOUR-SHAFT ARRANGEMENT.

greater in proportion to the power, and the blade heights in the annulus between the drum and the casing are relatively shorter; this gives a relatively greater clearance loss. The clearance area is practically constant for a given diameter, so that with higher pressures and denser steam the loss through clearance space will be greater.

Another point which has been greatly discussed in connection with turbine machinery is the question of the method of measuring the horsepower developed by the turbine engines. As is now more or less generally known, it is not possible to indicate the power of the turbine as in the ordinary method of the indicator diagram of reciprocating engines. In the absence of any such method, it is found most convenient to take the estimated I. H. P. as the equivalent indicated horsepower which would be required with reciprocating engines for the particular class of vessel proposed. Messrs. W. Denny & Bros. have recently introduced a new dynamometer or torsion meter for obtaining the power transmitted by the shaft, the readings being obtained by

the torsional deflection of a given calibrated length of the main shafting. This instrument has been used on several vessels, and appears to give very reliable results; and where facilities are at hand for obtaining the effective horsepower by means of tank experiments the shaft horsepower, together with the effective

Vessel.	Length.	Beam.	Depth.	Tons.	Speed.
Steam yacht:					
Turbinia (*a*)	100	9	7	45*	34.5
Emerald (*b*)	198	28.7	18.5		15.
Lorena	253	33.3	20.3		18.01
Tarantula	152.5	15.3	8.8		25.36
Albion	240	34	20		15.
Narcissus	210	27.5			14.5
Mahroussa	400	42			17.5
Destroyer:					
Cobra	223.5	20.5	13.5		34.6
Viper (*c*)	210	21	12.5		36.88
Velox	210	21	12.5		27.12
Eden	220	23.5	13.8		26.23
River steamer:					
King Edward (*d*)	250	30	17.8	562	20:48
Queen Alexandra (*e*)	270	32	18.8		21.63
Duchess of Argyll	230	30			20.
Kingfisher	275	32			20.
Channel steamer:					
The Queen (*f*)	310	40	25	1,676	21.73
Brighton (*g*)	280	34	22		21.37
Princess Maud (*h*)	300	40	24.5		20.66
Londonderry	330	42	25.3		22.29
Manxman (*i*)	330	43	18		23.
Viking (*j*)	350	42	16		23.53
Princesse Elisabeth (*k*)	344	40	23.2	1,747	24.03
Onward	310	40	25		22.53
Invicta	310	40	25		22.5
Dieppe (*l*)	284	34.7	22		21.5
St. David	350	41			22.5
St. Patrick	350	41			22.5
St. George	350	41			22.5
Viper (*m*)	315	39.5			21.75
Atalanta	210	27			17.5
Cruiser:					
Amethyst (*n*)	360	40		3,000*	23.63
Torpedo boat (French):					
293 (*o*)	130	14			26.2
Transatlantic Liner:					
Virginian (*p*)	520	60	41	10,630	19.11
Victorian (*p*)	520	60	41	10,630	18.75
Carmania (*q*)	678	72	52	19,524	20.
Ocean steamer:					
Loongana	300	43	16.5		20.15
Maheno (*r*)	400	50	38.5		16.5
Lhasa	275	44	25.5		18.09
Linga	275	44	25.5		18.05
Lama	275	44	25.5		18.
Lunka	275	44	25.5		18.
Rewa	455	56			16.5
Bingera	300	40	19		16.
Lake steamer:					
Turbinia (*s*)	250	33	20.8	1,100*	18.46

* Displacement.

See Marine Engineering (*a*) 6, June, 1897; 11, July, 1897; 14, Dec., 1897; 28, Feb., 1898; 26, March, 1898; 1, Jan'y, 1899. (*b*) 196, April, 1903; 317, June, 1903. (*c*) 213, Nov., 1899. (*d*) 395, Sept., 1901; 8, Jan'y, 1902. (*e*) 422, August, 1902. (*f*) 528, Oct., 1903. (*g*) 10, Jan'y, 1904. (*h*) 323, July, 1904. (*i*) 487, Dec., 1905. (*j*) 443, Nov., 1905. (*k*) 488, Dec., 1905. (*l*) 247, June, 1905. (*n*) 37, Jan'y, 1905. (*o*) 43, Jan'y, 1905. (*p*) 359, Sept., 1905. (*q*) 1, Jan'y, 1906. (*r*) 489, Dec., 1905. (*s*) 427, Sept., 1904; 5, Jan'y, 1905. (*m*) International Marine Engineering, 321, August, 1906.

horsepower, form a very good basis for estimating propeller efficiency.

Other advantages incidental to the turbine might be cited, such as reduced weight of machinery, steadiness and smoothness of running, reduction in oil and stores, and absence of racing in a seaway, and other advantages which have brought the turbine to a commercial success.

The development of the marine turbine has taken place almost entirely in Great Britain. A few war vessels have been built in France and Germany. The reason that the Parsons marine turbine has made so little progress on the continent in the past is probably due to the fact that in France and Germany rival turbine systems of local origin have been energetically exploited, inducing those responsible for the ordering of new vessels to defer the adoption of a new system until the claimants had results to show.

The Parsons turbine was introduced to the continent by the small French torpedo boat No. 293. This boat, ordered by the French navy as an experiment in the year 1903, has been in continual service for over two years without any breakdown or repair, and continues to give in practice the results obtained on trial. Naturally, so small a boat hardly demonstrates the results which might be expected from larger installations. The above mentioned experimental torpedo boat is 130 by 14 feet, and of 95 tons displacement. She attained a speed of 26.2 knots. The arrangement of machinery is similar to that in the *Turbinia*, three turbines in series.

In France, a mail and passenger boat is being built for the Marseilles and Algiers service of the Compagnie Generale Transatlantique. She is 378 by 43 feet, and is designed for a service speed of 20 knots. In the United States five passenger vessels are being built, and a scout cruiser of 24 knots speed for the government, to be fitted with Parsons turbines.

Reference to the table will show the vessels which have been engined in Great Britain with Parsons turbines up to the present date, representing a total horsepower actually completed of about 280,000. The total horsepower of marine turbine engines completed and on order with the Parsons Marine Company and their licensees is over 870,000.

Comparison Between Turbines and Reciprocating Engines.

The illustration shows in one view one of the steam turbines to be fitted in the new scout cruiser *Salem*, building in the yards of the Fore River Shipbuilding Company, Quincy, Mass., and one of the triple expansion reciprocating engines fitted in the battleship *Vermont*, building in the same yard. In each case the requirement is for about 8,000 horsepower, so that the comparison is direct.

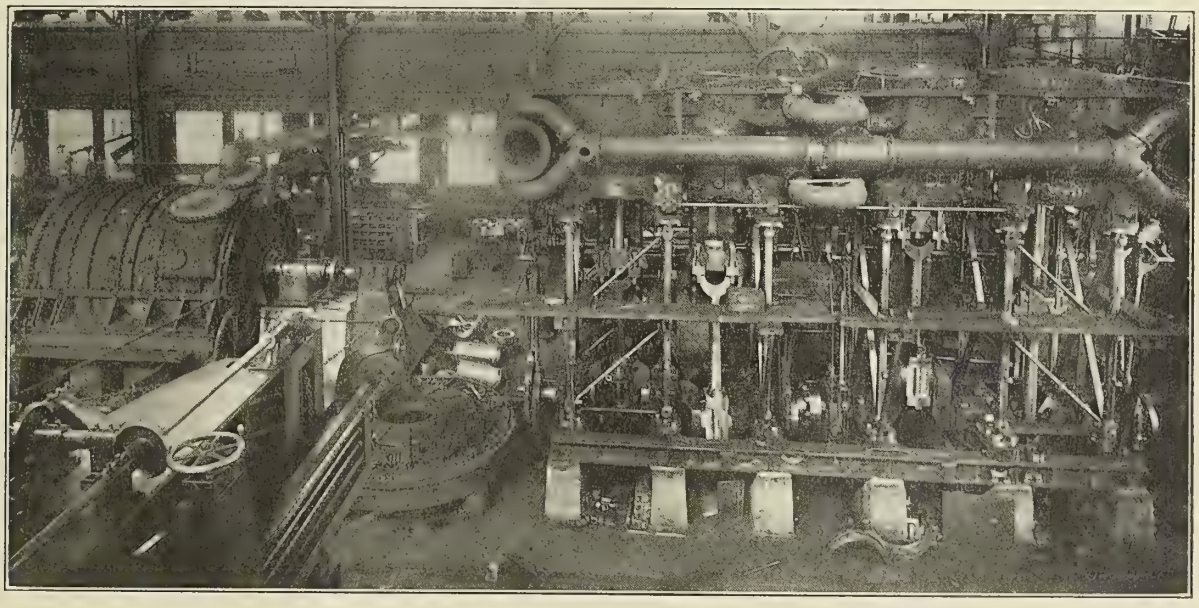
The turbine illustrated is identical in size and pattern with that illustrated in our October number in connection with the description of the new Southern Pacific steamer *Creole*, although it is of double the power of that turbine. This astonishing increase in power with the same turbine casing and rotor is obtained by the joint use of an increase in the number of revolutions and in the pressure of the steam, the former resulting from the latter. The number of nozzles admitting steam is also increased in the higher powered turbine.

The present turbine, which is of the 120-inch size, 7-stage, operates at 350 revolutions per minute and weighs 228,680 pounds, or a trifle more than 102 tons. This figures out at 28.6 pounds per horsepower. The reciprocating engine, which has four cylinders with diameters 32½, 53, 61 and 61 inches, and a common stroke of 48 inches, operates at 120 revolutions per minute, and weighs 343,350 pounds, or a little more than 153 tons, this being almost exactly 50 percent more than for the turbine. This weight figures out at about 43 pounds per indicated horsepower.

A comparison of the dimensions and space occupied by these two types of prime mover will be interesting.

	Turbine.	Engine.
Length over all	16 ft. 2¾ ins.	33 ft. 6½ ins.
Width over all	13 ft. 6 ins.	11 ft. 3 ins.
Height over all	12 ft. 6 ins.	21 ft. 9 ins.
Floor space	219 sq .feet.	377 sq. feet.
End area	169 sq. feet.	245 sq. feet.
Side area, or target	203 sq. feet	730 sq. feet.
Over all volume	2,735 cu. ft.	8,200 cu. feet.

In the above tables the areas and volumes are figured out from the gross dimensions in each case, and are, therefore, strictly comparable. The turbine shaft over all measures 23 feet 7 inches, while the crank shaft of the engine is 31 feet 1 inch. The length from center to center of main bearings of the turbine is 18 feet 6 inches, corresponding with 25 feet 8½ inches from center to center of the end bearings of the engine. The length over the stuffing boxes of the turbine is 14 feet 5½ inches. The length over all of the cylinders of the engine is 32 feet 9 inches. The width over all of the bedplate of the engine is 10 feet. The diameter of the flanges of the stuffing box casing of the turbine is 11 feet 2¾ inches. In the table the figures for the turbine are based on the over all dimensions of the casing.



ENGINE AND TURBINE, 8,000 HORSEPOWER EACH, IN SHOPS OF FORE RIVER SHIPBUILDING COMPANY.

The *Salem* is a scout cruiser measuring 420 feet between perpendiculars, with a beam of 47 feet 1 inch, a draft of 16 feet 9 inches and a displacement of 3,750 tons. A speed of 24 knots is expected on a total of 16,000 horsepower. The normal coal supply is 500 tons, which may be increased to a total of 1,250 tons with bunkers all filled. In the latter condition, and with a full supply of stores and ammunition on board, the displacement is 4,687 tons. The over all length is 423 feet 1 inch. The battery consists of twelve 3-inch rapid fire guns and two 21-inch submerged torpedo tubes. The contract price for hull and machinery was $1,556,000, the date of contract being May 17, 1905, and estimated date of completion March 17, 1908.

The *Vermont* is a first class battleship, measuring 450 feet between perpendiculars, with a beam of 76 feet 10 inches, a draft of 24 feet 6 inches and a displacement of 16,000 tons. A speed of 18 knots is expected with 16,500 horsepower. The normal coal supply of 900 tons can be increased to a maximum of 2,200, under which condition, with full stores and ammunition on board, the full load displacement is 17,650 tons. The over all length is 456 feet 4 inches.

The *Vermont* carries a battery of four 12-inch and eight 8-inch breech loading rifles mounted in pairs in turrets; twelve 7-inch rapid fire guns; twenty 3-inch and thirty smaller guns; and four 21-inch submerged torpedo tubes. The armor belt has a thickness of 9 inches; turrets and barbettes range from 12 inches down to 4 inches; the protective deck varies from 1½ to 3 inches. The contract price for hull and machinery was $4,179,000, the contract having been entered into on June 20, 1903, and calling for completion of vessel December 20, 1906. The keel was laid May 21, 1904, and the vessel launched August 31, 1905.

Society of Naval Architects and Marine Engineers.

Announcement is made of the program for the fourteenth general meeting of the Society of Naval Architects and Marine Engineers, which took place on Thursday and Friday, November 22 and 23, 1906, in the auditorium of the American Society of Mechanical Engineers at 12 West 31st street, New York City. A list of the papers read follows:

THURSDAY, NOVEMBER 22, 1906.

1. *A Fireproof Ferryboat,* F. L. DuBosque.
2. *Construction of a Fireproof Excursion Steamer,* William Gatewood.
3. *Speed and Power Performance of Albany Day Line Steamer NEW YORK,* Professor J. E. Denton.
4. *Personal Impressions of Model Towing Stations Abroad,* Professor C. H. Peabody.
5. *The Experimental Tank of the University of Michigan,* Professor H. C. Sadler.
6. *Model Basin Gleanings,* Naval Constructor D. W. Taylor, U. S. N.
7. *Comparison of the Durand and Curtis & Hewins Papers on Propeller Experiments of 1905,* Colonel E. A. Stevens.
8. *A Modern Fleet,* Captain E. B. Barry, U. S. N.
9. *Recent Developments in Armor and Armament,* John F. Meigs.
10. *The Development of War Ship Design,* Naval Constructor R. H. Robinson, U. S. N.
11. *Development of Submarines,* Lawrence Y. Spear.
12. *The Effect of the Universal Rule on Yacht Design,* Martin C. Erismann.
13. *The Gasoline Engine,* Arthur T. Chester.
14. *Two Timber Dry Docks and Their Pumping Plants,* F. P. Palen and G. L. Smith.
15. *A New Sea Anchor for Coaling at Sea,* Spencer Miller.

Following the meeting, a banquet was held at Delmonico's at 7 P. M. on Friday, the 23d of November. We expect to give in our January issue a full account of the meeting, with abstracts of the papers and the principal parts of the discussions.

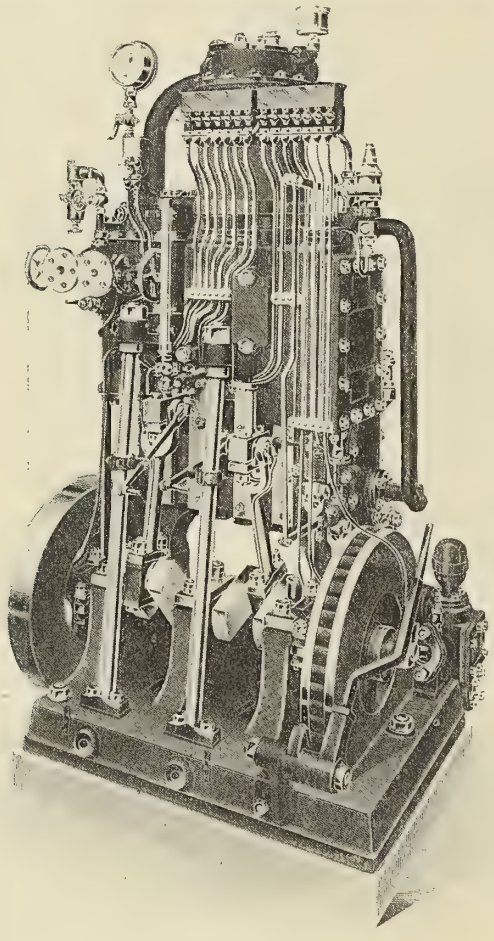
COLD STORAGE ON BOARD SHIP.

PART VI.

BY SYDNEY F. WALKER, M. I. E. E.

CIRCULATING PUMPS.

Two kinds of circulating pumps are required for refrigerating apparatus, apart from the compressor, which is itself a pump—that for the condensing water, and that for the brine, where brine is employed. The sizes of the pumps and their arrangement hardly require any explanation to marine engineers. They will depend upon the quantities of water and brine to be circulated, and the sizes and lengths of the pipes through which they pass. The pumps can be driven by any convenient source of power. With small plants it is usual to mount them on the outside of the tank which forms the condenser, and to drive them from



HASLAM NAVAL REFRIGERATING SET, MOUNTED IN TANDEM.

the compressor crank shaft. With larger plant, separate arrangements may be made. It must be remembered that the metal of the brine pump becomes very cold and that the moisture in the atmosphere condenses on it and freezes.

The work done by the heat delivered to the generator in the absorption apparatus is the equivalent of the work done by the engine driving the compressor, and the economy of the system, as opposed to that of compression, turns upon this point—the relative economy of using the steam direct, and using it to drive a steam engine which in its turn drives a compressor. The heat delivered to the generator must be sufficient to convert the liquid ammonia into gas, as well as to raise the temperature of the solution, so that the gas may be expelled. A proportion of this heat is obtained from the heat delivered by the gas to the water in the absorber, in the act of solution, and the further this can be carried, the more economical must the process be.

HOW REFRIGERATING APPARATUS IS MEASURED.

In a previous part of these articles, the terms one-ton machine and six-ton machine have been used. These terms express the manner in which refrigerating apparatus is measured. A one-ton machine is one that will produce what is called a ton of refrigeration in twenty-four hours. A ton of refrigeration is the equivalent of the cooling effect that the melting of a ton of ice would produce in the same time. Ice requires, as mentioned, the abstraction of 142 heat units for every pound of water that is frozen, from and at 32 degrees F. In melting, each pound of ice will absorb 142 heat units, and a ton of ice therefore will absorb, on American measurement, 284,000 units, and on British measurement, 318,080 units—America using the simpler short ton of 2,000 pounds, while the United Kingdom uses the old-fashioned 2,240 pounds. The above figures mean, respectively, 11,833 and 13,253 units per hour. It is important to remember this latter fact.

It may happen that it is required to extract heat very rapidly. The most rapid rate for a one-ton machine is given by the above figures, the capacity of larger or smaller machines being in direct proportion. It is also important to note that a one-ton machine will not make one ton of ice in twenty-four hours. It is usually reckoned that the quantity of ice which will be made by a machine of any given size is one-half its rating. The reason is, there are losses between the refrigerating plant and the water that is being frozen, added to which work has to be done in abstracting heat from the water in reducing it to freezing point, and from the ice after it is actually frozen. The old rule holds here as everywhere—do not cut your plant too fine; and another rule which is equally true—you can afford to work closer to the rated capacity of any plant, the greater care you give it.

Practice allows a one-ton machine for a cold chamber ranging from a capacity of 800 cubic feet up to 3,000 cubic feet, a two-ton machine from 3,000 to 6,000 cubic feet, a four-ton machine from 10,000 to 20,000 cubic feet, and so on.

THE POWER REQUIRED FOR REFRIGERATING APPARATUS.

The power required varies from 0.3 horsepower per ton of refrigeration, up to 1.7 horsepower. Small plants require more power in proportion than large plants, and more power is required, usually, in hot climates than in temperate. Small plants are allowed from one horsepower up to three horsepower for a one-ton machine, the proportion falling as the size is increased. The power required increases as the condenser pressure increases, and that is one reason why more power is required in hot climates. In addition, more power is required because more water has to be circulated, when its initial temperature is higher.

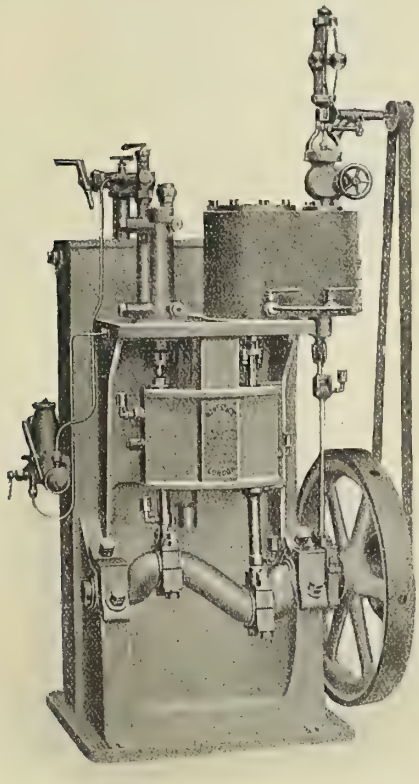
COOLING WATER.

A point that is of importance here is to note that if sea water is employed its specific heat is lower than that of pure water. It may easily have a specific heat as low as 0.95, which means that a little over 5 percent additional cooling water must be allowed. For cooling water having an initial temperature of 55 degrees F., and having its temperature raised to 80 degrees F., the following quantities of pure water are required: for the ammonia condenser, 50 gallons an hour per ton of refrigeration, and with carbonic acid about 15 percent more, according to the Linde Company's experiments; larger quantities are required, as explained, when the initial temperature is higher. Approximately, ten gallons per ton additional will be required for every five degrees increase in initial temperature, unless the final temperature can also be increased, and again larger quantities for carbonic acid.

FORMS OF APPARATUS FOR USE ON BOARD SHIP.

The apparatus that is most suitable for ship board use is the one that occupies the smallest space, provided other things are equal. Some forms of apparatus are made very compact, the casting upon which the compressor and engine are built inclosing

the condenser for the refrigerator, while the condenser for the steam engine which drives the compressor is carried overhead. The driving arrangements are also made very compact, the steam engine bed being extended at the back of the cylinder, and



WEST CARBONIC ANHYDRIDE COMPRESSOR AND CONDENSER.

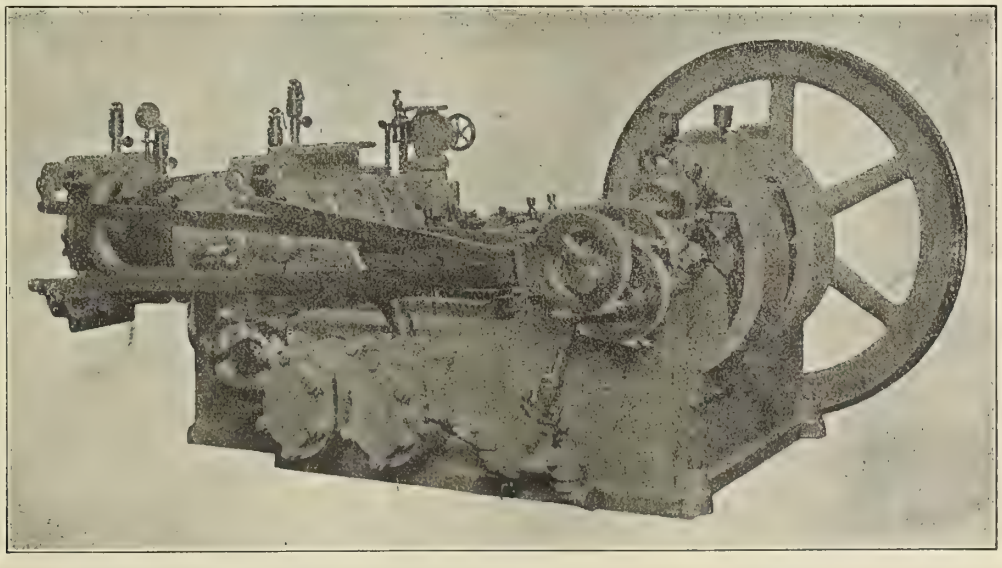
the tail rod of the piston being used as the piston rod of the compressor. A useful form of apparatus that has been fitted into some ships has two steam cylinders, mounted on one bed plate, each driving its own carbonic acid compressor by the tail rod of its piston as explained above, and each having its own steam and its own carbonic acid condenser.

are tested to a pressure several times that which they are likely to have to stand; but it is well to keep the spare bottles cool. With increased temperature the gas endeavors to expand, and if it is not able to do so, the pressure increases inside the bottles. With iron and steel it is a very difficult matter to be sure that there are no flaws, hidden away in the body of the metal, which the increased pressure would find. Also, stoppers sometimes become faulty after a time. In any case, it is a very simple matter to provide for keeping the bottles cool. A cupboard is built, large enough to take the spare bottles that are carried. It is insulated in the usual way, and has a small grid of pipes carrying the refrigerant or brine, as convenient; or it may be cooled by exposure to the current of air, or a branch of it, that is cooling the cold chamber.

COOLING MAGAZINES AND OFFICERS' AND MEN'S QUARTERS.

Trouble appears to have arisen in the British Navy over the matter of cooling the magazines. It was reported some time since that fans were put on to cool the magazines, but they produced trickling water on the walls, and so had to be abandoned. With refrigerating apparatus on board, the problem should be solved with ease. The air entering the magazine should be dried before its entrance, by one of the methods described. Probably a small grid and a small fan would do the work very well. But it must be remembered that for successful ventilation the air that is displaced must be disposed of. It is of no use trying to force fresh air into the magazine unless the air already in the magazine is carried away. There must be a complete ventilating circuit, just as there must be a complete electrical circuit, or no action can take place; but the atmosphere may form part of some ventilating circuits, just as the ground forms part of some electrical circuits.

The same remarks apply to cooling the officers' or men's quarters, or to cooling the 'tween decks, when live cattle are carried there. Putting a fan in somewhere and churning up the air will not ventilate. It may create drafts, but it will only move the air, not change it. A current of air brought down from the deck, and directed through the space to be cooled, will cool to the extent that its temperature and the quantity available allow, as explained in connection with the cooling of air for cold chambers. But where the atmospheric air is at a high temperature, especially if it is heavily charged with vapor, it has very little cooling effect. The figures given above will show this. If the air is cooled a few degrees, and dried, it will then produce a



STEAM-DRIVEN DRY AIR REFRIGERATING MACHINE, LIGHTFOOT SYSTEM.

OTHER APPLICATIONS OF REFRIGERATION ON BOARD SHIP.

One useful application of cold temperature is to keep low the temperature of the carbonic acid that is stored. The refrigerants employed are always carried in strong iron or steel bottles, which

pleasant, cooling effect on both man and beast, because, with men at any rate, nature's cooling apparatus, the evaporation of perspiration from the skin, can come into play. This evaporation can take place in a dry atmosphere, even if it is not a cool one.

SHORE END OF MUSKOGEE WHARF (L. & N. RY.) FROM WHICH AN ENGINE AND FORTY CARS OF COAL WERE LOST.

The Pensacola Hurricane.

During the night of September 26-27 a terrific West India hurricane devastated the country in the neighborhood of Pensacola, Florida, and in particular caused immense destruction to shipping in the harbor. Many ships of considerable size were beached or were damaged or pounded to pieces by striking one another after their cables had been broken or anchors dragged from the mooring grounds. Six vessels in the port proved total losses, while as many more which were driven ashore are in nearly as bad condition.

SLIP BETWEEN PALAFOX AND BALEN STREET WHARVES, PENSACOLA, FLORIDA.

The German bark *Marie,* 1,022 tons, which has been abandoned, was wrecked completely by being forced from her moorings, and pounded from the stern by the Norwegian bark *Campbell,* 1,100 tons, which was also wrecked. The *Marie* was forced upon the shore and the cargo, which had been partially put aboard her, has been removed. The *Campbell* is split in halves, having broken from her mooring at the east side of Tarragona dock.

WRECKED STEAM TUGS IN PENSACOLA HARBOR.

The Norwegian ship *Inquid,* 1,260 tons, which had just started discharging a full cargo of cement imported from Hamburg, was thrown high up on the rocks, and has been completely abandoned. The ship was under charter to reload for Rio de Janeiro.

The Portuguese ship *Ferreria,* 963 tons, which was originally a British ship engaged in the coolie trade, was another total loss. The ship was fully loaded and almost ready to sail. The Norwegian ship *Avanti,* 1,831 tons, was left in a very precarious position with small chance of being saved. She was loaded and bids were asked for the removal of the cargo.

Other vessels which were badly damaged but not destroyed

OVERTURNED FISHING SCHOONER AND WRECKED FREIGHT TRAIN.

NORWEGIAN BARK CAMPBELL AND GERMAN SHIP MARIE.

GENERAL AND TYPICAL VIEW OF WHARF WRECKAGE AFTER THE STORM.

include the British ship *Canada,* 2,237 tons; Italian ship *Lena,* 1,587 tons; Norwegian bark *Hereford,* 1,524 tons; Norwegian bark *Heimdal,* 1,491 tons; and the Uraguayan bark *Iliua,* 700 tons. Our illustrations show the sort of wreckage resulting from a storm of this magnitude.

THE USE OF KEROSENE OIL IN ENGINES BUILT FOR GASOLINE.

BY H. B. MAXWELL.

The constantly increasing demand for gasoline, owing to new fields for its use being rapidly developed, has caused a steady increase in price, which makes the use of the heavier and cheaper liquid hydrocarbon fuels a matter of such importance that it has for some time past attracted the attention of engineers engaged in designing internal combustion engines. There are a number of commercial engines on the market for stationary, and a few for marine, use that are regularly built to use kerosene, but in nearly all of them the designers and inventors, as is usually the case, have used a complicated mechanism to reach the desired end.

To digress a little from the subject in hand, attention may be called to the fact that gas naphtha, running from 58 to 62 degrees Beaumé, can be used successfully in any well designed four-cycle engine, and in two-cycle engines after they have been heated up by running a short time on gasoline. A few users of gasoline engines are aware of this fact, and where so situated as to obtain it use it exclusively, and as it costs only six or seven cents per gallon, the saving is quite an item.

To use gas naphtha successfully, note must be taken of the fact that, owing to its being heavier than gasoline, it takes more air to furnish the proper amount of oxygen for complete combustion. The vaporizer or carbureter should therefore be adjusted for the heavier fuel by closing the needle valve until a properly combustible mixture is found.

With any fuel, the cooling water in the cylinder jacket should be so restricted that it leaves the jacket at a temperature of from 180 to 200 degrees Fahrenheit; with air or oil cooling, higher temperatures are desirable; in fact, with any internal combustion motor using any kind of fuel the hotter it can be run without causing preignition, or destroying the lubricant, the greater the efficiency will be. It will surprise the average user of a gas or gasoline engine to make a few tests of fuel consumption with varying temperatures of the cooling water at the outlet, as it will be found that when the water circulation is so rapid as to keep the engine cold, an appreciably greater quantity of fuel will be used than will be the case where the water is raised nearly to the boiling point before leaving the jacket.

Kerosene can be used in many engines after being warmed up on gasoline, but not economically unless some device is added to insure perfect combustion of the fuel or to aid in the better vaporization of the fuel before it reaches the combustion chamber. It has been found that by using a common mixing valve of the type shown in Figs. 1 and 2, kerosene can be used successfully.

Referring to Fig. 1, it will be noticed that the nut forming the cap to the body of the valve is tapped for a priming cup. This is to be filled with gasoline, and the engine started on this. As soon as the engine is heated up a little, open the needle valve controlling the kerosene inlet. After doing this a blue smoke will be noticed at the exhaust pipe. Then turn on the valve controlling the water supply, a little at a time, until this smoke clears away, being sure that lubricating oil is not being used in excess. This will work on most four-cycle engines using high compression.

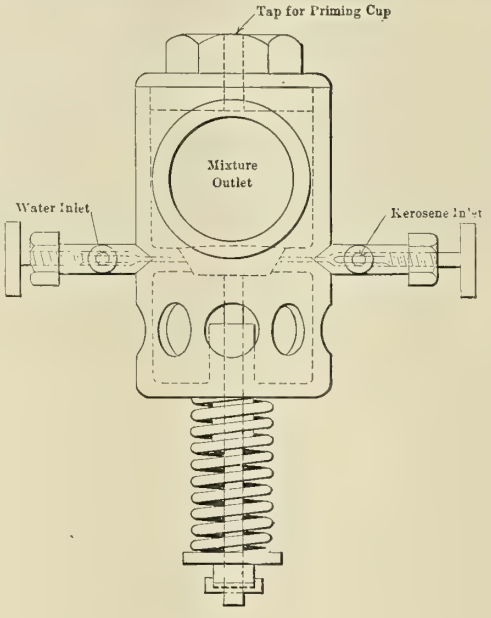


Fig. 1

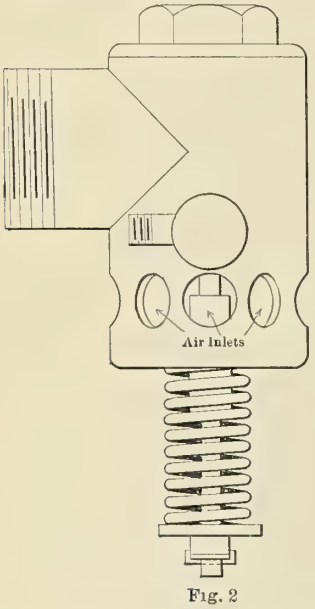


Fig. 2

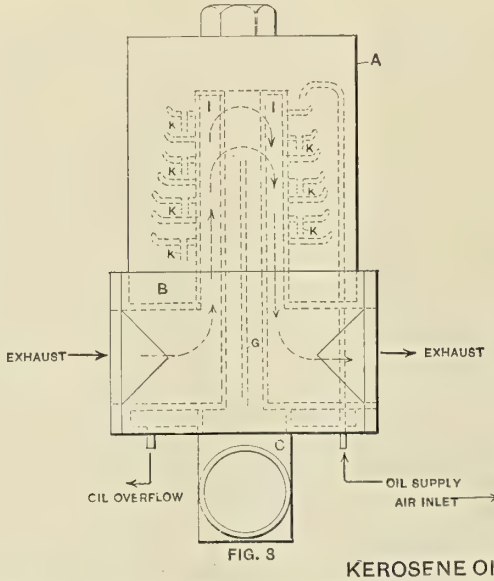


FIG. 3

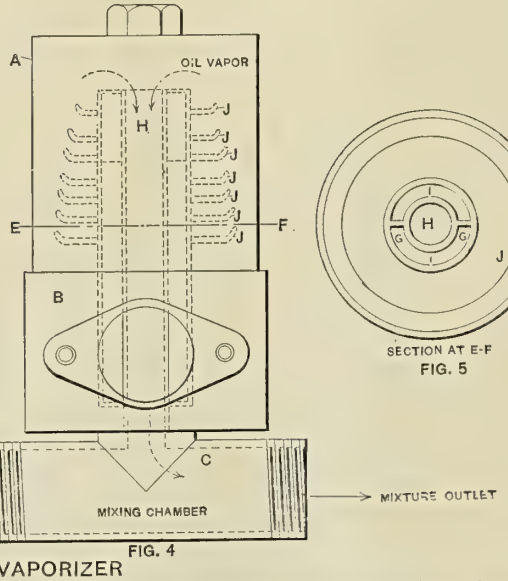
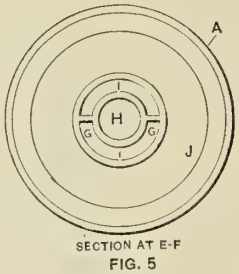


FIG. 4



SECTION AT E-F
FIG. 5

KEROSENE OIL VAPORIZER

Figs. 3, 4 and 5 show an exhaust heated vaporizer that can be used with any engine. It consists of a body *B* and *C,* cast integral with the pipes and oil pans, as shown in the dotted lines, Figs. 3 and 4, and section, Fig. 5. The cap *A* screws into the body *B,* and can be readily removed to clean the pans of any tar or coke deposited from the fuel used. The oil supply may be operated by a variable stroke pump controlled by a governor, or the engine can be controlled by throttling the mixture

between the vaporizer and the intake valve. There must be an air regulator attached to the mixture chamber *C* at the end opposite from the end used to convey the mixture to the engine.

The exhaust enters the T-shaped pipe *I I* and impinges against the pipe *H* and the guides *G G*, giving the exhaust gases the course shown by dotted arrows in Fig. 3. This imparts part of the heat of the exhaust gases to the walls of the pipe *I I* and the pans *J J*. It will be noticed that the overflow pipes *K K* from each pan to the one next below it bring the fuel not vaporized by the first hot surface into contact with other hot pans, to complete the vaporization. Any overflow is taken care of by the outlet at lower left hand side of Fig. 3. The oil, gas or vapor flows down pipe *H* to the chamber *C*, where it is mixed with the proper amount of air to furnish the correct mixture.

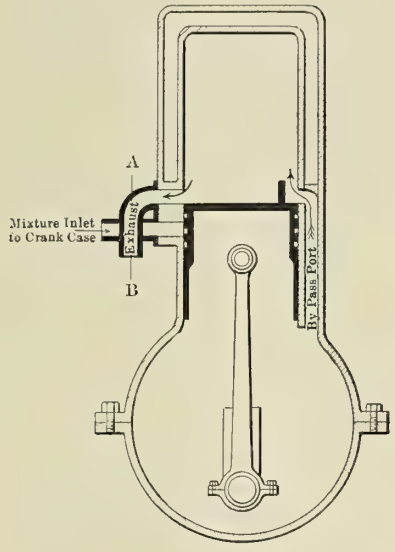


Fig. 6

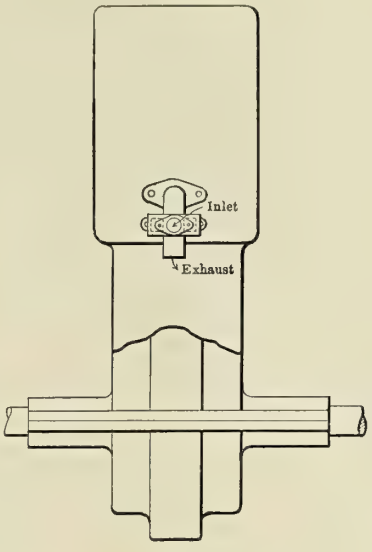
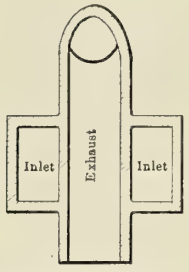


Fig. 7



Enlarged Section at A.B.
Fig. 8

Figs. 6, 7 and 8 show a device designed by the writer, and *not* patented. Referring to Fig. 6, it will be noticed that the exhaust and intake pipes are cast integral. The inlet from carbureter at the left opens directly against the hot walls of the exhaust pipe, and as shown in Fig. 8, divides and passes around the exhaust pipe and enters the port to the crank case.

This works so well that it is now being applied to a two-cycle air cooled engine, with every confidence that it will solve the problem for a cheap and reliable motor. Of course, to start a two-cycle engine with this device, it must be primed with gasoline, the priming cup being located on top of the cylinder; or the combination exhaust and inlet pipe could be heated with a torch.

The Fast Launch Svip.

Built and engined by John I. Thornycroft & Company, Limited, Chiswick, London, W., the *Svip* has a length of 35 feet, a beam of 5 feet 9 inches, and a draft of 1 foot 6 inches.

The boat is carvel built of cedar, bright varnished inside and out. The engines are two 24-horsepower 4-cylinder Thornycroft motors, driving twin screws, the ignition being the standard low-tension magneto. The exhaust is carried through a muffler fitted in a neat funnel, which is fixed to a baffle plate immediately over the engines.

The boat is intended for fast cruising, has comfortable seating accommodation for 12 to 16 people, and can be adapted to stand a very fair sea. A feature of the boat is the sliding hood, which can be collapsed and rolled up at the forward end when desired.

THE THORNYCROFT MOTOR BOAT SVIP, ON THE THAMES.

The boat attained a speed of 20¾ miles per hour on the first occasion on which the engines were started up, a performance which may be considered very satisfactory. The builders guaranteed the boat (under a penalty) to be capable of a speed of 20.5 miles per hour with two persons on board. On the official trial the boat ran six times over the measured mile, with and against the tide, at an average speed of 21.2 miles per hour, three persons being carried. It will be seen, therefore, that the boat exceeded its contract speed by nearly 0.7 mile per hour, with an extra passenger on board.

Personal.

Frederick A. Goetze has been appointed dean of the School of Applied Science of Columbia University, to take effect January 1.

A NEW TORSION METER, FOR MEASURING THE POWER OF MARINE STEAM TURBINES.

BY FRANK C. PERKINS.

Torsion meters are particularly valuable in steam turbine work, as it is impossible with this form of motive power to take indicator cards, as with reciprocating engines. By means of this instrument the power developed and transmitted from a steam turbine can be ascertained, and accuracy of 99 percent obtained in the readings. The instrument has been utilized to great advantage on a number of the turbine steamers, the recording box being placed in the cabin where there is very little noise and the sound of the telephone receiver is distinctly heard. Recording machines have also been devised and installed with a moving strip of paper, a pen which reports the number of revolutions by means of electric contacts, another pen connected with the bridge so as to take the time on the mile, and a third pen marking the half seconds from a standard chronometer.

The accompanying illustrations indicate the construction and arrangement of the new torsion indicator constructed at Glasgow, Scotland, by Kelvin and James White, Limited, after designs of Charles Henry Johnson, and Archibald Denny. One view shows one of the instruments installed on a turbine steamer, the disk being fixed at one end of the center shaft; the disk fixed to the remaining shafts and to the other end of this shaft are similar. Another illustration shows the recording box, while a third shows two inductors of a somewhat different construction of torsion meter, the indicator *A* consisting of a thin soft iron core wound with insulated wire, and arranged on a base provided for the purpose, while the inductor *B* is a similar core wound with insulated wire, and mounted on a slider on a quadrant-shaped guide in such a way that by turning a handwheel the slider carrying the core may be moved along the guide either forward or backward as desired.

The telephone receiver is placed in a circuit with the inductors, which are placed beneath a wheel carrying a magnet. In order to take a reading the handwheel is moved until silence is obtained in the telephone receiver. This adjustment is noted, and indicates the amount of torsion of the shaft. This form of torsion meter is intended for low speeds, and can be utilized to advantage for measuring the torsion of shafts of vertical or horizontal recriprocating engines.

On account of the uneven turning moment of steam engines, the torsion of this kind of shafting varies during a single revolution. For this reason six magnets are utilized on each wheel instead of one, these magnets being equidistant from the circumference of the wheel, and on a different parallel plane, while six separate windings are arranged on each conductor, and a switch is provided by means of which any pair of windings on the two inductors can be simultaneously connected in the circuit with the receiver.

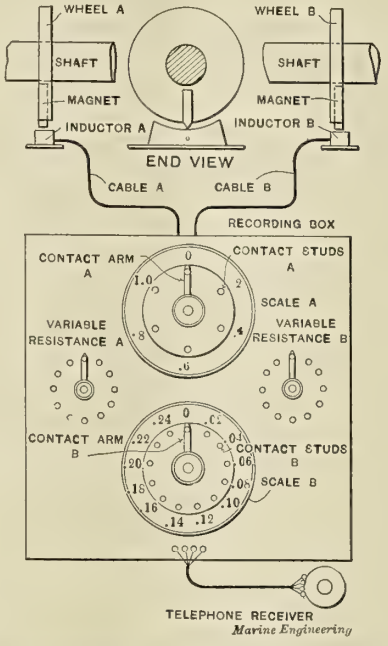


FIG. 1.—DIAGRAM OF THE ELECTRIC CIRCUIT.

Resistances are arranged for adjusting the strength of the induced current, and in this manner the torsion at six different points can be measured easily, and therefrom the mean torsion for complete revolution can be noted. It may be stated that on a test with a reciprocating engine which had a turning moment of considerable variation it was noted that with six magnets the greatest error was not over 4 percent, and when working out a polar diagram of torque the error was not more than 1 percent, with the magnets suitably arranged on the wheel.

One of the special features noticed in this instrument is the absence of rubbing contact, thus avoiding wear and tear. These instruments are particularly well adapted for work on shipboard, making it possible to take frequent readings of the power, and note with accuracy any loss of efficiency. For testing steam

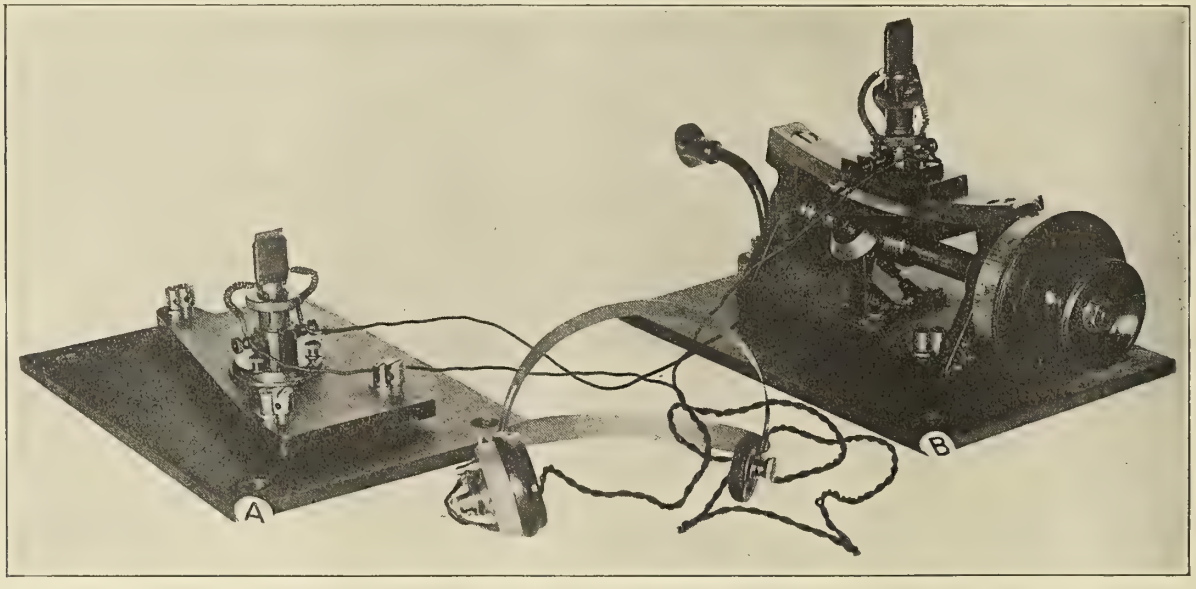


FIG. 2.—THE TELEPHONIC RECEIVING MECHANISM OF THE TORSION METER.

turbines they are of great value, and have given excellent satisfaction.

The diagram shows the arrangement of the torsion meter when connected with turbine driven shafting. Two light wheels, *A* and *B*, similar to that shown in Fig. 1, are fastened to the shaft at proper distances apart, and a permanent magnet mounted on each wheel in such a manner that the projecting pole produces a dense and definite magnetic field at the point, by being constructed in the shape of a *V*. There are two inductors, *A* and *B*, set concentrically with the wheels and shaft, and just below the magnets. Each of these inductors consists of a quadrant-shaped piece of soft iron mounted on a gun metal stand provided with leveling screws. A number of separate windings are arranged on each piece of iron, a suitable number of turns per unit of circumferential length of iron being used.

A recording box of the type shown in Fig. 3 is used, having two series of contact studs, *a* and *b*, which are marked with proper scales, and two contact arms *A* and *B*, so that electrical connection can be made with any stud of a series and with its contact arm, there being in series *A* a stud for every separate winding in the inductor *A*, and similarly for every separate

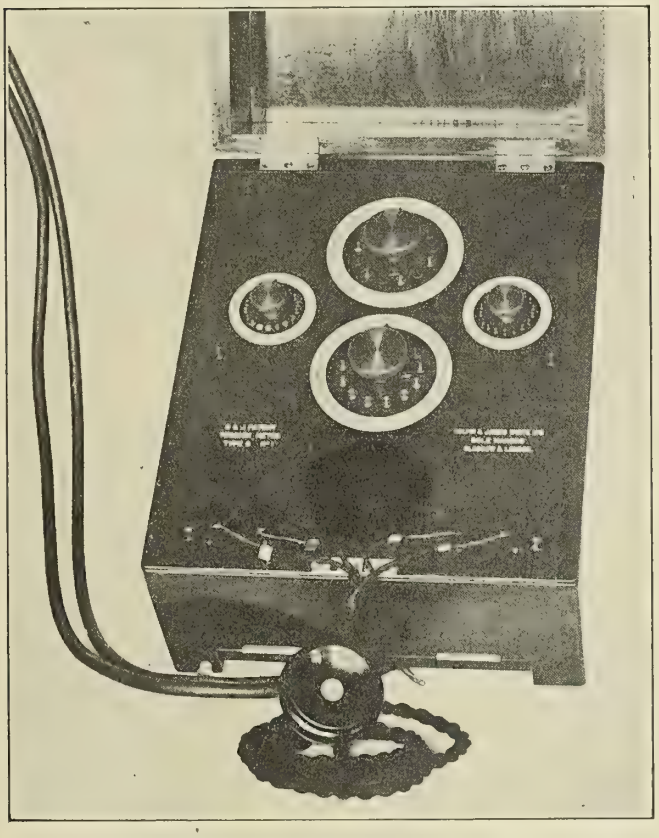


FIG. 3.—RECORDING BOX SHOWN IN FIG. 1.

winding in the inductor *B* there is a stud in series *B* of the recording box. Each stud is connected to its own particular winding by means of an independent wire, and all the wires are contained in the multiple cables *B* and *A* connected with the inductors *B* and *A* respectively. There is a variable resistance provided for each of these two circuits, and one winding of a differential telephone receiver. There are six equal parts and six separate windings in the inductor *A*, and the scale *A* is therefore divided into six parts, there being six studs in the *A* series. It will be noted that the length of the five subdivisions of the scale represents the circumferential length occupied by all the windings on the inductor, this being about 0.2 inch, each subdivision being the distance between windings.

There are fourteen equal parts to the scale *B*, as there are fourteen independent windings on this inductor, and in the series *B* fourteen studs, with a distance between windings of 0.02 inch. The inductor at the turbine end of the shaft has its wheel so set that its magnet is above one or the other of the two end windings in the end inductor. When the shaft rotates without transmitting any power a current of electricity is generated in the end winding of each inductor, and these two separate currents traverse their own circuits in passing from the inductor winding to the studs, and then through the contact

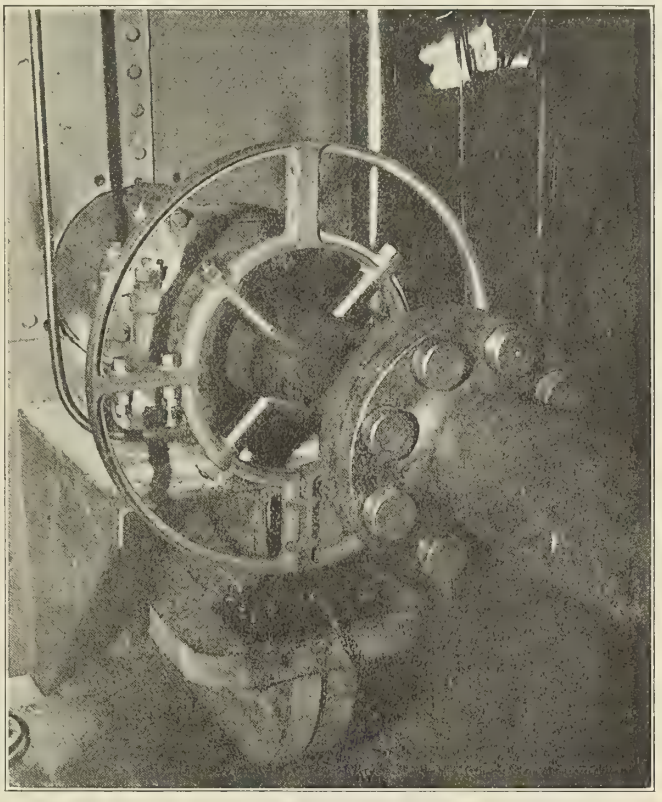


FIG. 4.—ANGLE MEASURING INSTRUMENT ON SHAFT.

arms and resistances, as well as to the telephone receivers and in returning to the inductors.

The receiver windings are so connected that the effects of the independent currents are in opposition, and neutralize each other at the receiver when the two current strengths are the same. With the variable resistances the currents can be made equal in strength, and in this manner when the shaft transmits no power and there is no torsion there will be no sound in the receiver. As soon as the shaft begins to transmit power, and there is a twist or torsion, there will be a loud ticking sound noted in the receiver, as the currents no longer neutralize each other. It is then necessary to move the contact arm *B* from one stud to another until silence is again obtained in the receiver. By noting the reading opposite the scale *B*, the angle of torsion of the shaft will be noted. The scale reading represents the displacement of one magnet with regard to the other, due to the torsion of the shaft.

The National Association of Engine and Boat Manufacturers announces the National Motor Boat Show to be held in Madison Square Garden, New York, from February 19 to 26 inclusive. It was decided last year that this show was to be exclusively a motor boat show, the sportsmen element being divorced from the motor boat element, and being the subject of a separate show. It was also decided, on account of the great demands made upon the space within the garden, and particularly upon the main floor, to dispense with the tank, which has been so prominent a feature of the last two shows. Arrangements are made for the operation by illuminating gas of internal combustion motors.

Launch of a Floating Dock.

The following is a general description of the new floating dock built by Messrs. Vickers, Sons and Maxim, Limited, for the British Admiralty, for use at Haslar, and which was launched recently.

The over all dimensions of the dock are: length, 250 feet; width, 50 feet 6 inches; depth of pontoon at center, 6 feet 4 inches; length of side walls, 200 feet; height of side walls above lower edge of pontoon, 22 feet 6 inches; height of side walls and pontoon, 28 feet. The hull is constructed throughout of open bearth mild Siemens Martin steel.

The dock is required to lift a vessel displacing 500 tons and of a maximum draught of 10 feet within a space of one hour. The dock, however, has a net lifting power of 1,000 tons, and is capable of docking a vessel having a draught of 16 feet. In the center of the starboard wall, a two-storied motor house with galleries is fitted, to contain on the ground floor the electric motors and starting resistances, on the second floor immediately above, the valve levers and switch board. A three-ton hand crane mounted on a traveling under carriage is placed on the starboard wall, and on each wall of the dock an electric capstan is fitted. In addition to the usual keel blocks, three double mechanical side shores are provided in each wall. The pumping gear is worked by electric motors. The dock is lighted throughout by electricity, and is fitted up with all the usual appliances.

NEW FLOATING DOCK FOR SUBMARINES; PORTSMOUTH HARBOR, ENGLAND.

Developments in Warship Design.

At the eleventh annual meeting of Sir W. G. Armstrong, Whitworth & Company of Newcastle-on-Tyne, England, which was held September 28, 1906, the chairman of the Board of Directors called attention to recent battleships and armored cruisers designed in England, the main object having been to increase the power of the armament and, at the same time, to simplify its application. In the most recent designs there have been placed only two calibers of guns, aside from machine guns, consisting of powerful armor-piercing weapons and of quick firing guns of greater power than formerly used, and fitted for the purpose of keeping away hostile torpedo craft. The principal armament in recent designs consists of from 8 to 16 armor-piercing guns, and there are indications that the company in question would engage to increase the number to 20, with a corresponding increase in the displacement of the vessel. The secondary armament in vessels designed at this works was placed at from 16 to 24 quick firing guns of 4-inch or 4.7-inch caliber, the number and size depending upon the size and character of the vessel under construction.

Another question of very great importance is that of speed. The old advantage of the weather-gage has taken a new form known as speed-gage. Each of these items enables the possessor to accept or decline combat at will, depending upon the particular situation existing at the moment. The use of turbine machinery has enabled shipbuilders to give to warships of great power a higher speed than has heretofore been the case, and thus to obtain all the advantages inherent in superior speed.

Steamer for Cuba.

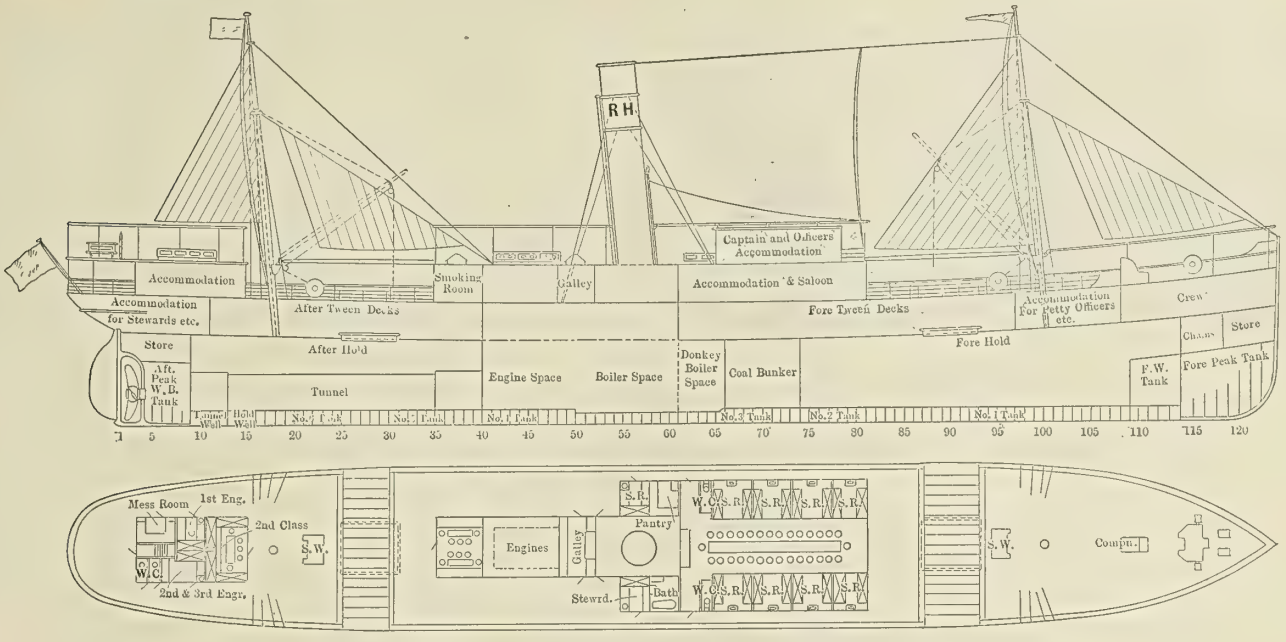
An interesting vessel for Havana has been launched from the shipbuilding yard of William Hamilton & Company, Limited, Port Glasgow, in the shape of a steel screw passenger and cargo vessel of about 1,450 tons gross, built to the order of Sobrinos de Herrara Societe en la Ciudad de Habana, for their Cuban coastal trade. This vessel is 235 feet long, 35 feet beam and has a depth of 24 feet to the awning deck; and she has been built under special survey of the British Corporation for their highest class.

Accommodation is provided for 33 first-class passengers at each side of a large teak deckhouse placed on the awning deck forward of the boiler casing. The cabin enters off the dining saloon, which is situated between the two rows of staterooms. The dining saloon is arranged to seat 33 persons, and is tastefully decorated, well lighted from skylights overhead, and ventilated by means of electric fans. A handsome smoking room is fixed aft of the engine casing.

Second-class accommodation is arranged in the fore end of a large deckhouse on the aft end of the awning deck, and accommodation for captain, officers and engineers is provided at the after part of the second-class accommodation. In a deckhouse on top of saloon house deck are the chartroom and steering house. The accommodation for crew is forward under the awning deck, and for servants, stewards, etc., at the after end under the awning deck.

The 'tween deck is lighted throughout by sidelights and is sheathed with pitch pine so that it may be used for carrying cattle. The awning deck is of teak. For the quick handling of cargo, each hatch is provided with a powerful steam winch and three derricks. The vessel is lighted throughout by electricity, and has a powerful steam windlass and steering gear.

The *Habana* is designed to steam 11½ knots in service, and is fitted with triple expansion surface condensing engines constructed by Muir and Houston, of Glasgow, with cylinders 19, 31 and 51 inches in diameter and stroke of 36 inches; and two boilers 14



INBOARD PROFILE AND ACCOMMODATION PLAN OF NEW CUBAN STEAMER HABANA.

feet diameter by 10 feet 6 inches long, and 180 pounds working pressure, natural draft. BENJAMIN TAYLOR.

A Motor Torpedo Boat.

The original idea in the construction of fleets of torpedo boats was the provision of a mosquito squadron consisting of large numbers of small and inexpensive vessels, exposing a small area to gunfire, and having a high speed to enable them to successfully attack larger vessels. Developments, however, particularly in the line of increased speed, have gradually brought about an increase in size, until the latest vessels of the torpedo type, or destroyer class, displace nearly one thousand tons, as compared with forty or fifty tons in the earlier types. As the cost has increased more than proportionately with the displacement, the number of vessels built has correspondingly decreased, and it has long been a question whether better protective results might not have been attained with a larger number of smaller vessels.

To meet this condition, Messrs. Yarrow & Company have brought out a torpedo boat propelled by internal combustion engines operating three propellers. The boat is 60 feet long with a beam of 9 feet, and displaces only 8 tons, which makes it readily transportable by rail from one point to another, thus following what might be called an "inside route" of communication. The structure is very light, and the engines are of sufficient power to give the boat a speed of 24 knots when carrying a load of 3 tons in the shape of torpedoes and fuel. The latter is carried in a special tank which does not form any part of the structure of the hull, while the armament may consist of either one torpedo in a revolving tube or two in drop gear at the side.

THE YARROW MOTOR-DRIVEN TORPEDO BOAT ON HER TRIALS.

It is estimated that a steam driven boat of the same size, and carrying the same load, could be provided with machinery capable of giving her a speed of only 18 knots. It is, moreover, estimated that the radius of action of such a steam-driven boat, on one ton of fuel, would be sixty miles, as compared with three hundred miles for the present craft. Vibration is said to be very slight, even at full speed of the engine (1,000 revolutions per minute). It is stated that three hundred and fifty of these vessels could be built for the price of one *Dreadnought*.

INTERNATIONAL MARINE ENGINEERING

Published Monthly at
17 Battery Place - - New York
By MARINE ENGINEERING, INCORPORATED
Christopher St., Finsbury Square, London, E. C.

TERMS OF SUBSCRIPTION

	Per Year	Per Copy
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Other countries in the Postal Union	10/6	1/

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The edition of this issue comprises 6,000 copies. We have no free list, accept no return copies and issue only enough to supply the regular demand.

Marine Steam Turbines.

The subject of the steam turbine as applied to marine uses is becoming one of ever increasing importance. Our present number contains an unusually large quantity of material on this subject, including the reprint of a paper read by Mr. Parsons himself; accounts of the trials of the British battleship *Dreadnought,* equipped with four Parsons turbines; a description of the German merchant steamer *Kaiser,* equipped with Curtis turbines; a comparison between Curtis marine turbines and reciprocating engines; and a description of an instrument and method for measuring the power of steam turbines when applied to ship propulsion.

The wonderful flexibility of this instrument of propulsion was briefly mentioned in our comment last month in connection with the powering of the new British battleships of the *Invincible* class, in which it is expected to attain an extremely high speed, with a large margin of power. This flexibility receives further notice in connection with the Curtis turbine illustrated on page 481 of this issue, in the comparison there made. Practically the only difference between that turbine of eight thousand horsepower and the smaller one of four thousand horsepower applied to the Southern Pacific steamer *Creole* is in the number of nozzles through which steam is admitted, the pressure of the steam at admission, and the resulting number of revolutions per minute of the shaft. The rotors and casings of the two turbines are identical, not only in design but in dimensions, yet in the one case, due to a difference in the conditions of use of the steam, the power is exactly double that in the other.

It is difficult to know in any given case just what power may be expected of such an instrument. So much depends upon the boiler equipment, and more particularly upon the capacity of the boilers to furnish any quantity of steam which may be utilized by the prime mover, that the ultimate power seems to be partially independent of the size of the latter. The Editor remembers having tested a steam turbine rated at two hundred horsepower; but which, on being given a larger amount of steam than that for which it was designed, was operated at as much as 350 horsepower. This is right in line with the relation between the turbines of the *Salem* and those of the *Creole.* It also serves to illustrate the point brought out last month, in connection with the possibility of developing in the *Invincible* a power exceeding by ten thousand horsepower that for which the turbines have been designed.

Heretofore, tests of marine steam turbines have been figured out on a basis, a large proportion of which was estimate or approximation. This was due to the fact that the turbine is not fitted for the taking of indicator cards such as may be obtained from any piston engine. Devices now in use, however, such as that described on pages 488 and 489, afford a ready means of obtaining accurate readings of the power actually transmitted by the turbine. This method depends, of course, upon the constancy of the elasticity of the material of which the shafting is composed, but for all engineering purposes this elasticity is sufficiently constant to make the indications of the torsion meter thoroughly reliable to a degree of accuracy beyond which it is unnecessary to go.

One point in which indications of this sort differ from those obtained by the indicator cards of a reciprocating engine is, that the latter show the entire power exerted by the engine, both in overcoming its own internal and external friction, and in doing the useful work for which it was designed. With the turbine indications, on the other hand, the internal friction of the machine is absolutely eliminated, and the power, as measured, is the net power transmitted to the shaft for the purpose of doing useful work. The difference between the two is one of varying magnitude, some engines having been sufficiently well designed, and well operated, to reduce the engine friction to a figure as low as three or four percent. It is unusual, however, to get this below seven percent, and frequently from twelve to fifteen percent has to be accepted as a working condition. There is thus no direct comparison possible between figures given for a reciprocating engine tested in the usual manner, and those obtained from the indications of a torsion meter as applied to the steam turbine engaged in marine propulsion. When a reciprocating engine, however, is tested through the intermediation of an absorption or transmission brake, or dynamometer, the results so

obtained are strictly comparable with those obtained from the torsion meter.

In comparing coal or steam consumption figures, as between the two types of prime movers, account must necessarily be taken of this fact of the discrepancy in the results, and proper allowance made for the difference. For instance, the fuel consumption of the *Dreadnought* at full power figured out at 1.51 pounds per horsepower per hour. This, it will be noted, is based on the developed horsepower and, assuming 9 percent engine friction for the purpose of illustration, the coal consumption per "indicated" horsepower per hour would be only 1.36 pounds. This is a figure which will be recognized to be extremely good, and one which very few steam engines of any type have ever surpassed. In the designing of machinery for ocean liners and freighters, it is usual to assume that the conditions will be such as to call for an expenditure of 1.6 or 1.7 pounds of coal per indicated horsepower per hour, and this under conditions most favorable to the development of power at the greatest economy. In warship practice it is not at all unusual for the consumtion to figure out above 2 pounds per horsepower hour, and in many cases it is more than 2.5 pounds. These comparisons will suffice to illustrate the excellence of the results obtained upon the *Dreadnought.*

The number of turbine propelled vessels is increasing very rapidly, not only in the United States and in England, but also on the continent of Europe. It is fortunate, for the interests of marine engineering and naval architecture in general, that developments along this line are proceeding from several points of view at once. In England the only type of turbine used in this development is the Parsons. On the continent of Europe several types are being developed, the Rateau in France, the Riedler-Stumpf and Curtis in Germany, and a modification of the Parsons turbine also in Germany.

In the United States, the Curtis turbine is distinctively the American type, but the developments of the Parsons type are being fitted to vessels by two separate concerns working along independent lines, and achieving results in different ways. It may be said that neither of these developments of the Parsons turbine is exactly the same as the developments taking place in England; although, of course, both are founded upon the same basic principles, and the general outline of the results is identical. It is in the small innovations and alterations in the design and construction of such a device that the strength of the future lies with regard to the ultimate development of the system, and it is for this reason that we count it most fortunate to have this development proceeding from the minds of so many different engineers at the same time.

Internal Combustion Engines.

Following up a development of the motor boat engine in connection with the use in that engine of some fuel other than gasoline, we presented in our July number a paper covering the use of denaturized alcohol in the explosive engine, showing what changes were necessary in the design and operation of the engines in order to make the change in fuel effective. It is found that the principal change in the construction is in the carburetor, where the mixing takes place between the fuel vapor and the air with which it is diluted before being admitted to the cylinder. The use of alcohol has been found to require in the engine some pre-heating of the cylinder before operation becomes successful. This is usually done by means of a priming charge of gasoline, upon which the engine is operated for a few revolutions, or until the cylinder becomes sufficiently heated to make the alcohol take care of its work in proper manner.

In another column we present another paper, covering the use of kerosene in this same form of prime mover, and it has been found that the same conditions governing the use of alcohol in a gasoline engine obtain with respect to the use of kerosene in the same engine.

The price of gasoline has been increasing with the steadily growing demand for it for fuel purposes, due to the large multiplication of engines in motor boats and automobiles. It is largely with a view to obtaining a cheaper fuel, or, at any rate, to keeping the price of the fuel down to the lowest possible point, that the developments have proceeded, looking to the utilization of other fuels in the explosion engine. It is for this reason that the two papers above mentioned are especially valuable to engineers engaged in the designing or operation of such engines, and it is expected that, as developments in design proceed, it will become easy to change an engine from any one of these fuels to any other, without delay or stoppage due to making the transition. This state of affairs is much to be hoped for, because it will largely increase the sphere of influence of the internal combustion engine, and render it a much more flexible instrument for the production of power than is at present the case.

The Naval Architects' Meeting.

Seven groups of papers are included in the list of fifteen read at the 1906 convention of the Society of Naval Architects and Marine Engineers, there being four papers on model experimental tanks and their application; four on developments in warship design; two on fireproof steamers; two on yacht design and powering; and one each on speed trials, coaling at sea and dry-dock pumping plants. All of the papers are of importance and interest, and should command the attention of all students of marine construction, be they engaged in naval or mercantile work. But best of all was the opportunity presented by the meeting for experts to "get together" and discuss the various papers, bringing out the strong points of each, and submitting the subjects presented to a rigid and illuminating analysis. It is this contact of intellect which furnishes the highest value of such a society. The papers could well be printed in any place, but the discussions require in general the concurrent presence of numbers of master minds.

ABSTRACT FROM RULES FOR RIVETING NAVAL VESSELS.

BUREAU OF CONSTRUCTION AND REPAIR U. S. NAVY DEPARTMENT.

ARRANGED BY HARRISON S. TAFT.

1. All rivet holes through material 1 inch or more in thickness should be drilled, or, if punched, should afterwards be reamed to finished size. The increase in diameter of hole due to reaming should be equal to at least one-eighth the thickness of the material. In punching, where possible, holes will be punched from the side which will form the faying surface.

2. Great care must be taken in punching to prevent holes from coming unfair. Any unfair hole must be reamed out before riveting, and a rivet suitable to the increased size of hole inserted.

3. Counter sinking must be carefully done, following the rules for angle of countersink given on sheet I. Holes for countersunk heads and points will be countersunk through the plate or shape to a depth corresponding as nearly as possible to the dimension "B" given on plate 1 for the size and type of rivet to be used. In cases where such a depth would extend through the plate, the countersink should be carried to within 1/32 of the bottom.

4. Rivets less than ⅜-inch diameter should be driven cold. In torpedo-boat work rivets up to 7/16-inch, inclusive, may be driven cold, at the discretion of the Superintending Naval Constructor. Power riveting is to be preferred for torpedo-boat work.

5. Type of Rivets.

1. Wherever practicable, the pan-head rivet will be used, with countersunk points where flush work is required, button or snap points for finished appearance or where rivets are closed by power, and hammered or mashed points elsewhere. Where points are made with a snap tool, or where riveting by power is employed, care will be taken that the points are properly centered.

2. Care must be taken that snap points are not reduced from the standard sizes by grinding down tools that have been chipped or burred. All pan-head rivets not less than ½-inch diameter for punched holes should be coned under head, as shown on sheet 1, the rule for punching from the faying surface of plate being carefully observed. If, however, the practice of punching the holes small and reaming to size by power be employed, the coning under head may be omitted.

6. Diameter of Rivets.

TABLE 1. DIAMETER OF RIVETS.

	Weight of Plates in lbs.	Diameters	
		Rivets	Rivet Holes
Ship Work.	Up to 3 lbs. exclusive	1/4	9/32
	3 lbs. to 6 lbs. exclusive	3/8	7/16
	6 lbs. inclusive to 8 lbs. exclusive	1/2	9/16
	8 lbs. inclusive to 13 lbs. exclusive	5/8	11/16
	13 lbs. inclusive to 20 lbs. exclusive	3/4	13/16
	20 lbs. inclusive to 30 lbs. exclusive	7/8	15/16
	30 lbs. inclusive to 40 lbs. exclusive	1	1 1/16
	40 lbs. inclusive to 51 lbs. exclusive	1 1/8	1 7/32
	51 lbs. and above	1 1/4	1 11/32
Torpedo Boat Work	Up to 3 lbs. exclusive	1/4	9/32
	3 lbs. to 6 lbs. exclusive	5/16	11/32
	6 lbs. to 7½ lbs. exclusive	3/8	7/16
	7½ lbs. to 9 lbs. exclusive	7/16	1/2
	9 lbs. to 11 lbs. exclusive	1/2	9/16
	11 lbs. to 13 lbs. exclusive	5/8	11/16

NOTE.—The sizes of flanges of angles to which plates are connected in torpedo boat work may sometimes be such as to require a reduction in the size of rivet to secure satisfactory workmanship.

In cases where rivets connect plates of different thicknesses together, the size of rivet indicated for the greater thickness, with corresponding spacing, will be used where strength is required, and that indicated for the lesser thickness where water-tightness is a special consideration, always provided that the greater thickness is not more than double the lesser.

7. Tap Rivets.

Where tap rivets must be used they should be ⅛-inch larger than the corresponding ordinary rivets for the same thickness, excepting taps into heavy forgings or castings, such as stem and stern posts, which should be ¼-inch larger. Where strength is required tap rivets must penetrate *not less* than 1 diameter, and should penetrate 1½ diameters where the thickness of metal will allow it.

8. Spacing of Rivets.

1. Where the spacing given in Table No. 2 can not be followed exactly, as will generally be the case, the aim will be to make the spacing a trifle closer (as necessary in the case of heavier plating) and a trifle farther apart (as necessary in the case of lighter plating), the division between "lighter" and "heavier" plating coming at about 7½-pound plates for single riveting, at about 15-pound plates for double riveting, and at about 25-pound plates for treble riveting.

2. Where the above distinctions are considered too complicated for yard work, the general rule will be to space a trifle closer, in all cases, as necessary for equal spacing.

TABLE 2.—SPACING OF RIVETS.

Connection.	Number of diameters from center to center.
Single-riveted seams and butts	3½
Double-riveted butt laps and butt straps	4
Treble-riveted butt laps	4½
Treble-riveted butt straps, with alternate rivets in third row omitted	4
Longitudinal seams of plating required to be water-tight (excepting single-riveted seams specified in item 1)	4½
Connections of transverse frames not water-tight to outside plating	8
Connections of deck plating to beams; of non-water-tight longitudinals to outside plating; of the angles and stiffeners to bulkheads when entirely above the water line, and in general where special strength is not required	8
Connections of floor plates, brackets, lightened intercostals, etc., to clips and angles; of the vertical keel angles to the flat and vertical keel plates and to the flat keelson plates beyond the limits of double bottom, provided water-tightness is not required	7
Connections of angles and other stiffeners to bulkheads at or below the water line; of boiler and engine bearings and foundations generally	6
Connections of inner-bottom plating to all frames and longitudinals	5
Connections of angles of water-tight frames and longitudinals to all plating, and in general where water-tightness is required between shapes and plates	5
Angles and other stiffeners to bulkheads forming supports to turrets, barbettes, connections of armor-shelf angles to plating, etc.	5
Connections between staple angles of water-tight floors and the floor plates	4½
In special cases of intercostals, beam ends, etc., where strength is required in connections of limited extent, and in all other exceptional cases, spacing to be as required by circumstances, except that the rivets in the same line should never be spaced less than	3

9. Type of Riveting.

When strength is required in laps and butted connections of plating, with the spacing indicated above, single riveting is suitable only for plating under 12½ pounds and double riveting for plating under 25 pounds. For maximum strength in connections of plating above 30 pounds it will generally be found that quadruple riveting is required.

10. Single Straps.

Single butt straps and edge strips, when single or double riveted, should be of the same thickness as the plates connected, and where the plates connected are of different thickness they should be of the same thickness as the lighter plate. Single butt straps, when treble-riveted, should be 1¼ times the thickness of plates which they connect.

11. Double Butt Straps.

Double butt straps should not be used for water-tight work, owing to difficulty in calking. They may be used to advantage, however, in connections requiring great strength but not water-tightness. The thickness of each strap should be one-half the thickness of plates connected for double riveted straps, and five-eighths the thickness for treble riveted straps. The spacing of rivets in rows should be calculated. For double butt straps the size of rivet to be used should be as follows:

For plates from 15 to 20 pounds, exclusive, ⅝-inch rivets.
For plates from 20 to 25 pounds, inclusive, ¾-inch rivets.
For plates above 25 pounds, as per Table 1.

12. Distance Between Rows.

Centers of rivets should be placed not less than 1⅝ times the diameter from the edges of plates connected. In double and treble riveting, for laps and single straps, the distance from center to center of rows, should not be less than 2½ diameters; in butt laps and double butt straps the distance between centers of rows should not be less than three diameters (butt laps should be at least double riveted). For zigzag riveting the distance between centers of rows should not be less than 1¾ diameters for rivets spaced four diameters apart in rows.

13. Length of Rivets.

Special care will be taken in riveting that rivets used are of sufficient length to insure a proper point, the aim being to have the rivet a trifle long,, if possible. Such cutting as is necessary should be done while the rivet is still a dull red, and the point finished after further cooling. Allowances for length over the thicknesses connected are given in Table 3 below, the allowance being for two thicknesses only. An additional allowance of 1/16 should be added to that given for each additional thickness connected, unless the additional thickness is less than 3/16-inch, when 1/32-inch additional allowance should be sufficient. The alowances given in the table are based upon the employment of hand riveting, and are not sufficient in the case of power riveting, for which an added allowance—about ⅛-inch—should be made in each case. This table must be used judiciously and not absolutely depended upon.

TABLE 3.—ALLOWANCE IN LENGTH OF RIVETS FOR POINTS.

Diameter of Rivet. Inches	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼
	Allowance for point, over two thicknesses connected. Inches.								
Type of Point;									
Countersunk	5/16	⅜	½	⅝	¾	¾	⅞	1	1 1/16
Hammered	⅜	7/16	½	⅝	¾	⅞	⅞	1	1 1/16
Button	..	..	½	⅝	¾	⅞	⅞	..	..

The allowances given above apply only to rivets which fit the holes neatly, as hereinbefore described.

TABLE 4. REDUCTION OF DIAMETERS TO INCHES.

Diameter of Rivet.	Number of Diameters.																		
	1 5/8	2 1/2	3	3 1/4	3 1/2	4	4 1/2	5	5 3/4	6	6 1/4	6 1/2	7	8	8 1/4	9 1/4	11 1/2	12 1/2	16 1/2	18 1/2
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
1/4 inch.....	3/8	5/8	3/4	13/16	7/8	1	1 1/8	1 1/4	1 7/16	1 1/2	1 5/8	1 5/8	1 3/4	2	2 1/16	2 5/16	2 7/8	3 1/8	4 1/8	4 5/8
5/16 inch.....	1/2	3/4	15/16	1	1 1/16	1 1/4	1 3/8	1 5/8	1 13/16	1 7/8	1 15/16	2	2 1/16	2 1/2	2 9/16	2 7/8	3 9/16	3 7/8	5 1/8	5 3/4
3/8 inch.....	5/8	15/16	1 1/8	1 3/16	1 5/16	1 1/2	1 11/16	1 7/8	2 1/16	2 1/4	2 5/16	2 7/16	2 5/8	3	3 1/16	3 7/16	4 5/16	4 11/16	6 3/16	6 15/16
7/16 inch.....	11/16	1 1/16	1 5/16	1 7/16	1 1/2	1 3/4	2	2 3/16	2 1/2	2 5/8	2 3/4	2 7/8	3 1/16	3 1/2	3 5/8	4 1/16	5	5 7/16	7 3/16	8 1/16
1/2 inch.....	13/16	1 1/4	1 1/2	1 5/8	1 3/4	2	2 1/4	2 1/2	2 7/8	3	3 1/8	3 1/4	3 1/2	4	4 1/8	4 5/8	5 3/4	6 1/4	8 1/4	9 1/4
5/8 inch.....	1	1 9/16	1 7/8	2	2 3/16	2 1/2	2 13/16	3 1/8	3 5/8	3 3/4	3 15/16	4 1/16	4 3/8	5	5 3/16	5 13/16	7 3/16	7 13/16	10 5/16	11 9/16
3/4 inch.....	1 3/16	1 7/8	2 1/4	2 7/16	2 5/8	3	3 3/8	3 3/4	4 5/16	4 1/2	4 3/4	4 7/8	5 1/4	6	6 3/16	6 15/16	8 5/8	9 3/8	12 3/8	13 7/8
7/8 inch.....	1 7/16	2 3/16	2 5/8	2 13/16	3 1/16	3 1/2	3 15/16	4 3/8	5	5 1/4	5 1/2	5 3/4	6 1/8	7	7 3/16	8 1/16	10 1/16	10 15/16	14 7/16	16 3/16
1 inch.....	1 5/8	2 1/2	3	3 1/4	3 1/2	4	4 1/2	5	5 3/4	6	6 1/4	6 1/2	7	8	8 1/4	9 1/4	11 1/2	12 1/2	16 1/2	18 1/2
1 1/8 inches....	1 13/16	2 13/16	3 3/8	3 5/8	3 15/16	4 1/2	5 1/16	5 5/8	6 7/16	6 3/4	7	7 5/16	7 7/8	9	9 1/4	10 3/8	12 15/16	14 1/16	18 9/16	20 13/16
1 1/4 inches....	2	3 1/8	3 3/4	4 1/16	4 3/8	5	5 5/8	6 1/4	7 3/16	7 1/2	7 13/16	8 1/16	8 3/4	10	10 5/16	11 9/16	14 3/8	15 5/8	20 5/8	23 1/8

TABLE 5. BUREAU OF CONSTRUCTION AND REPAIRS, U. S. NAVY. WIDTH OF BUTT LAPS AND BUTT STRAPS. SPACING OF RIVETS.

Class of Work		Diametral Pitch	Ordinary Ship.									Torpedo Boat.					
			Up to 3 ex.	3 in. to 6 ex.*	6 in. to 8 ex.	8 in. to 13 ex.	13 in. to 20 ex.	20 in. to 30 ex.	30 in. to 40 ex.	40 in. to 51 ex.	51 and above.	Up to 3 ex.	3 in. to 6 ex.	6 in. to 7½ ex.	7½ in. to 9 ex.	9 in. to 11 ex.	11 in. to 13 ex.
Weight of Plates. Lbs.																	
Diameter of Rivets. Inches.			1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1/4	5/16	3/8	7/16	1/2	5/8
Diameter of Rivet Hole. "			9/32	7/16	9/16	11/16	13/16	15/16	1 1/16	1 7/32	1 11/32	9/32	11/32	7/16	1/2	9/16	11/16
Widths of	Butt Laps.																
	Single riveted	3 1/4	1 3/16	1 5/16	1 5/8	2	2 7/16	—	—	—	—	1 3/16	1	1 1/16	1 7/16	1 5/8	2
	Double "	6 1/4	1 15/16	2 1/16	3 1/16	3 15/16	4 3/4	5 1/2	6 1/4	7	7 13/16	1 15/16	1 15/16	2 5/16	2 3/4	3 1/16	3 15/16
	Treble "	8 1/4	2 1/16	3 1/16	4 1/16	5 3/16	6 3/16	7 1/16	8 1/4	9 1/4	10 5/16	2 1/16	2 1/16	3 1/16	3 5/16	4 1/16	5 3/16
	Treble " Shell Plating	9 1/4	2 5/16	3 1/16	4 5/16	5 13/16	6 15/16	8 1/16	9 1/4	10 3/8	11 5/16	2 5/16	2 7/8	3 1/16	4 1/16	4 5/8	5 13/16
	Butt Straps.																
	Single riveted	6 1/2	1 5/8	2 1/16	3 1/4	4 1/16	4 7/8	5 3/4	6 1/2	7 5/16	8 1/8	1 5/8	2	2 1/16	2 7/8	3 1/4	4 1/16
	Double " Single Straps	11 1/2	2 7/8	4 1/16	5 3/4	7 1/16	8 3/8	10 1/16	11 1/2	12 15/16	14 3/8	2 7/8	3 9/16	4 1/16	5	5 3/4	7 3/16
	Double " Double "	12 1/2	3 1/16	4 1/16	6 1/4	7 13/16	9 3/16	10 15/16	12 1/2	14 1/16	15 5/8	3 1/16	3 7/8	4 11/16	5 7/16	6 1/4	7 13/16
	Treble " Single Straps	16 1/2	4 1/16	6 1/16	8 1/4	10 5/16	12 3/8	14 7/16	16 1/2	18 9/16	20 5/8	4 1/16	5 1/16	6 3/16	7 1/16	8 1/4	10 5/16
	Treble " Double "	18 1/2	4 5/8	6 15/16	9 1/4	11 9/16	13 7/8	16 1/8	18 1/2	20 13/16	23 1/8	4 5/8	5 3/4	6 15/16	8 1/16	9 1/4	11 9/16
	Seam Laps.																
	Single riveted	3 1/4	1 3/16	1 5/16	1 5/8	2	2 7/16	—	—	—	—	1 3/16	1	1 1/16	1 7/16	1 5/8	2
	Double " (a) Zigzag	5	1 1/4	1 7/8	2 1/2	3 1/16	3 3/4	4 3/8	5	—	—	1 1/4	1 9/16	1 7/8	2 3/16	2 1/2	3 1/16
	Double " Chain	5 3/4	1 7/16	2 1/8	2 7/8	3 5/8	4 5/16	5	5 3/4	—	—	1 7/16	1 13/16	2 1/8	2 1/2	2 7/8	3 5/8
	Edge Strips.																
	Single riveted	6 1/2	1 5/8	2 1/16	3 1/4	4 1/16	4 7/8	5 3/4	6 1/2	7 5/16	8 1/8	1 5/8	2	2 1/16	2 7/8	3 1/4	4 1/16
	Double "	11 1/2	2 7/8	4 1/16	5 3/4	7 1/16	8 3/8	10 1/16	11 1/2	12 15/16	14 3/8	2 7/8	3 9/16	4 1/16	5	5 3/4	7 3/16
	Treble "	16 1/2	4 1/8	6 1/16	8 1/4	10 5/16	12 3/8	14 7/16	16 1/2	18 9/16	20 5/8	4 1/16	5 1/16	6 3/16	7 1/16	8 1/4	10 5/16
Spacing of Rivets in inches.	1. Center of rivet from edge of plate		1 5/8	3/8	5/8	13/16	1	1 1/16	1 1/8	1 5/16	2	3/8	1/2	5/8	11/16	13/16	1
	2. Center to center of rows. a) Zigzag riveting for rivets spaced 4 dias. apart		1 3/4	7/16	5/8	7/8	1 1/16	1 5/16	1 1/2	1 3/4	2	2 3/16	7/16	9/16	5/8	3/4	7/8	1 1/16
	b) Laps and single straps, double and treble riveted		2 1/2	5/8	15/16	1 1/4	1 9/16	1 7/8	2 3/16	2 1/2	2 13/16	3 1/8	5/8	3/4	15/16	1 1/16	1 1/4	1 9/16
	c) Butt laps and double butt straps		3	3/4	1 1/16	1 1/2	1 7/8	2 1/4	2 5/8	3	3 3/8	3 3/4	3/4	15/16	1 1/16	1 5/16	1 1/2	1 7/8
	3. Single riveted seams and butts		3 1/2	7/8	1 1/16	1 3/4	2 3/16	2 5/8	3 1/16	3 1/2	3 15/16	4 3/8	7/8	1 1/16	1 5/16	1 1/2	1 3/4	2 3/16
	4. Double riveted butt laps and butt straps; treble riveted butt straps with alternate rivets in third row omitted		4	1	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2	5	1	1 1/4	1 1/2	1 3/4	2	2 1/2
	5. Treble riveted butt laps; longitudinal seams of plating required to be watertight (excepting single riveted seams, as per No. 3); watertight floor staple work		4 1/2	1 1/8	1 11/16	2 1/4	2 13/16	3 3/8	3 15/16	4 1/2	5 1/16	5 5/8	1 1/8	1 3/8	1 11/16	2	2 1/4	2 13/16
	6. Inner bottom plating to floors and longitudinals; watertight frames and longitudinals to all plating and watertight work in general; stiffeners and other angles on bulkhead, forming support to turrets, barbettes and connection of armor-shelf angles to plating, etc.		5	1 1/4	1 7/8	2 1/2	3 1/16	3 3/4	4 3/8	5	5 5/8	6 1/4	1 1/4	1 9/16	1 7/8	2 3/16	2 1/2	3 1/8
	7. Bulkhead stiffeners at or below the waterline; angles of boiler and engine foundations and foundations in general		6	1 1/2	2 1/4	3	3 3/4	4 1/2	5 1/4	6	6 3/4	7 1/2	1 1/2	1 7/8	2 1/4	2 5/8	3	3 3/4
	8. Clips and angles of floor plates, brackets, lightened intercostals, etc.; keelson angles beyond limits of double bottom, if not watertight		7	1 3/4	2 5/8	3 1/2	4 3/8	5 1/4	6 1/8	7	7 7/8	8 3/4	1 3/4	2 3/16	2 5/8	3 1/16	3 1/2	4 3/8
	9. Non-watertight frames and longitudinals to shell plating; deck plating to beams; bulkhead stiffeners above water line and in general where special strength is not required		8	2	3	4	5	6	7	8	9	10	2	2 1/4	3	3 1/2	4	5

a) and Zigzag Riveting in general. — Chain Riveting in all cases unless otherwise specified. * in. = inclusive; ex. = exclusive.

TABLE 6.—RIVETING OF ANGLES.

Flange Size.....Inches.....	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3½	4	4½
Diameter of Rivet..... "	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	⅞	1	1⅛
Center of Rivet from Heel of Bar..... "	9/16	11/16	13/16	15/16	1⅛	1¼	1⅜	1½	1¾	2	2¼	2½

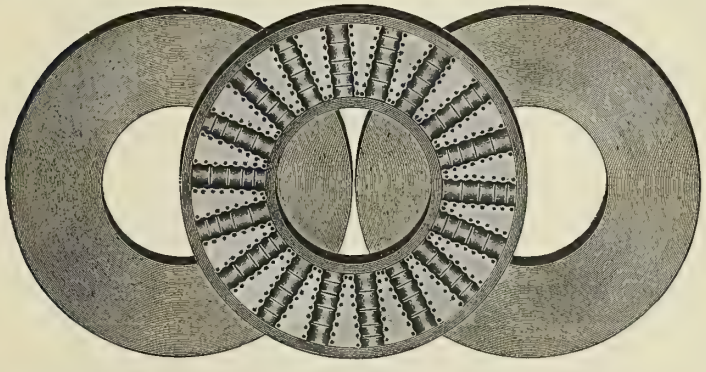
ENGINEERING SPECIALTIES.

The Contraflo Condenser.

This condenser, which was illustrated and described at some length in our September number, is manufactured by Messrs. Richardson, Westgarth & Company, Limited, Hartlepool Engine Works, Hartlepool, England. In principle, the direction of flow of the steam is opposed to that of the cooling water, which fact accounts for the name: contra-flow.

A Roller Bearing for Thrust Purposes.

Often there is need for a thrust bearing to eliminate friction upon shafting, cranes, turn tables, etc., where the space is too small to permit the use of a ball bearing, and for this purpose the Bantam Anti-Friction Company, Bantam, Conn., has produced a thrust roller bearing consisting of cylindrical rollers held together in a bronze or steel cage, as shown in the illustration,



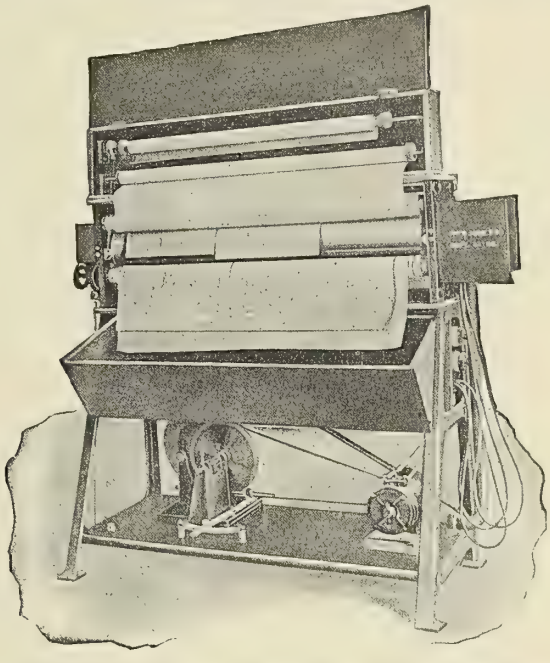
supported on either surface by hardened and carefully ground steel collars. The length of each roller is about one-half its diameter, which reduces to a minimum the slipping and sliding of the roller when revolving around the center of the shaft. For slow speeds these bearings give excellent satisfaction.

It is plain to every mechanic and engineer that the tendency of these rollers is to travel always at a tangent to the motion of the retaining case. At any high speed their action would be that of a milling cutter cutting endwise, and they would also slip and slide instead of rolling, but for slow speeds they give excellent results, and will carry enormous loads as compared with the ball bearing. They will eliminate about one-half of the friction found where such devices are not used.

The Everett-McAdam Blue Print Machine.

The machine consists of a rotating glass cylinder which lies in a series of narrow belts, and within which cylinder are placed two mercury vapor electric lamps. The roll of paper to be printed is placed in a box on top of the machine and feeds in continuously between the belts and the cylinders; or if only a few prints are wanted, previously cut sheets of paper may be fed in. The tracings are inserted between the paper and the cylinder, and, after passing around three-fourths of the circumference of the cylinder, are deposited with the paper in a box in the front part of the machine, the printing being done from the inside of the cylinder as the paper and tracings travel around it.

It is possible to make prints five feet wide and of any length whatever, and the device is particularly adapted to making numbers of small prints on one long sheet of paper, or on previously cut sheets—which come out in a regular stream—and of an absolutely uniform tone. One great feature of this continuous machine is that it suffices to handle one tracing at a time, the small ones being fed in side by side, while the paper feeds in automatically from a continuous roll. The tracings go in and come out on the same side of the machine, so that the leading edge of a tracing may be started into the machine again before the trailing edge comes out, saving considerable time where more than one print is wanted from one tracing. This was very nicely illustrated by



making twelve prints from one six-foot tracing on a single sheet of paper seventy-two feet long, without wasting an inch of paper or a second of time between prints.

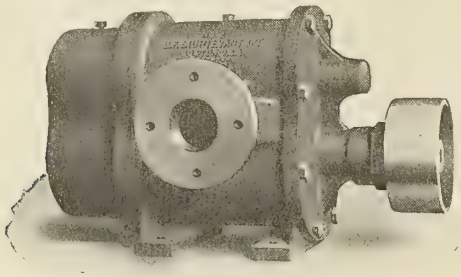
The lamps used are particularly adapted to this kind of work, as they give out only chemical rays—no energy being lost in non-chemical light, and as the printing is done from the inside of the cylinder, the light strikes the paper at right angles and the machine has an absolutely maximum electrical efficiency. The machine is very compact, requiring a space only two feet by five feet and is entirely self-contained. It is driven by a motor, and the speed can be instantly changed to any rate desired by moving a convenient lever. The machine has been placed on the market by the Revolute Machine Company, 523 West 45th street, New York.

High Pressure Blowers.

These are designed for handling air or gas at any pressure below ten pounds per square inch. They may be used to advantage in supplying air blast for foundry cupolas, forge fires, smelting furnaces, hardening, tempering, and annealing furnaces, whether supplied with coal, gas or oil; for pneumatic tube systems; for moving granular material; and in fact for furnishing air or gas for any purpose where the maximum pressure does not exceed the limit of ten pounds.

The blower, which is of the so-called positive type, is manufactured in a variety of sizes and capacities ranging from five cubic feet to over 15,000 cubic feet per minute. It consists of four principal parts; the cast iron shell or casing, two rotating members,

or rotors, and a stationary core. The rotors are held in position by bearings mounted on the end plates, and are made to revolve at the same speed, being connected by gears. As one of the rotors, called the impeller, revolves, it forms pockets or chambers in the annular space between the core and the shell, which imprison the air and carry it from the suction to the discharge, its return being prevented by the other rotor. Each pocket, as it nears the discharge, is decreased in volume, and consequently delivers its air at greater density.

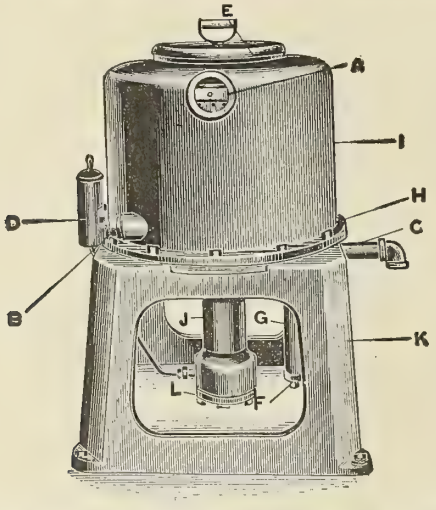


The other rotor, called the idler because it does no work, simply acts as a valve to allow the impeller blades to return to the suction side of the machine without loss of compressed air. All the work of compressing and moving the air is done by the impeller, which is mounted on the driving shaft, and no power is transmitted through the gears to the idler except that which is necessary to overcome the friction of the journals. Because all the work is done by one shaft, the load on the engine or motor is constant, the gears are relieved of all but a small fraction of their usual duty, and the wear is correspondingly reduced.

These blowers are built by the B. F. Sturtevant Company, Hyde Park, Mass.

Oil and Waste Saving Machines.

There has been installed on some of the battleships of the United States navy a machine for cleaning the filter cartridges used about the engine rooms, the cartridges being put into the machine and the oil or grease extracted. The machine is manufactured by the Oil & Waste Saving Machine Co., Philadelphia. It consists of a shell and a steam turbine driven basket. There

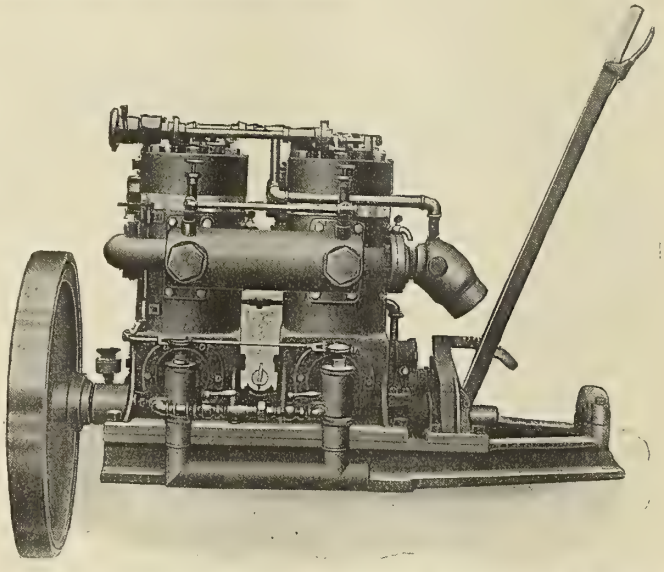


being no belts, shafts or other driving gearing, it readily appeals to the marine engineer. The material containing the oil or grease is put into the basket and the steam turned on. The steam passing through the jet strikes the turbine, revolving the basket, and then passes up through the material and exhausts at the top. The heat from the exhaust steam, acting on the oil or grease, loosens it and thus allows almost perfect extraction by centrifugal force. It takes from five to thirty minutes to a charge. In this time ninety-eight percent of the oil or grease is extracted from the waste or other material containing it. There being no foreign material used to aid extraction, the oil or grease and waste or other material are in condition to be used over and over again. The machine requires little space and can be set on the floor or hung to a wall or staging.

A Convertible Kerosene Motor.

We show herewith a convertible gasoline or kerosene motor which is being placed on the market by the Fay & Bowen Engine Co., Geneva, N. Y. This company has manufactured this motor for a number of months, during which time it has been thoroughly tested and perfected, the best evidence of its success being that their foreign agents, to whom the first machines of this type were shipped last season, are ordering them for this season in larger quantities.

The machine is a complete gasoline motor, and can be operated on gasoline. When used for kerosene the fuel is applied through needle valves above the exhaust pipe on the port side. The kerosene is placed under a slight pressure in its tank, not more than ten pounds pressure being required, and the operation and regulation of the machine under its kerosene fuel is said to be perfect. The charge is not ignited by the hot tube, which is the usual method in kerosene engines, but it is ignited by an electric spark, thus giving a vastly greater range of speed control than is usual in a kerosene motor.



When operated on kerosene exclusively the motor is started by removing two hexagonal plugs in the exhaust pipe, and applying heat from a torch for three to five minutes, when the machine will start readily. After being started and warmed up, it will stand for some time if necessary to stop it, and will start again without further heating. If it is not desirable to use a torch, the machine can be started by priming the cylinders with gasoline from a squirt-can, through the priming cocks on the cylinder heads, and after priming two or three times the machine is sufficiently warmed to operate on the kerosene fuel. It can also be started by the use of gasoline in the regular way through its gasoline vaporizers, and after starting, the gasoline can be shut off and the kerosene turned on, when the motor will operate entirely on its kerosene fuel. These different methods of starting give the operator a wide range, and make the machine practical for either the pleasure launch or the heavy service boat.

In this machine the vaporization of the kerosene fuel is very thorough and the unpleasant odor from the exhaust, which has heretofore been a disagreeable feature of all kerosene machines, is largely eliminated. The manufacturers of this motor have also conducted a series of experiments with alcohol, and the machine is found to operate successfully on that fuel, applied and handled in the same manner as kerosene. The motor is being made in single or multi-cylinder types, from 3½ horsepower upwards.

TECHNICAL PUBLICATIONS.

Parsons Turbine Engineer's Log. Size, 11 by 13 ¼ inches. Pages, 32. Glasgow, 1906. James Brown & Son.

This consists of a log for sixteen days, giving all the particulars required by the engineer through the six watches of the day, and covering the various items included in the engineer's log for a transatlantic or other liner. The thing is worked out in considerable detail, and spaces are given for various kinds of fuel, oil and engine stores, as well as for temperatures, weather observations, power indications by torsion meter, revolutions, steam pressures, etc. The book should be very handy for all engineers operating turbine machinery, because everything necessary to be recorded day by day appears to be provided for in the tables.

The Theory of the Steam Turbine. By Alexander Jude. Size, 6¼ by 8¾ inches. Pages, 296; illustrations, 252 + three folding plates. Philadelphia, 1906. The J. B. Lippincott Company. Price, $5.00 net.

In this volume an attempt is made to present the fundamental principles of the operation of a steam turbine in a concise and connected manner, and in a way which will enable these principles to be applied directly to the turbine problem. The descriptive matter included has been curtailed to a minimum, but is in sufficient quantity to afford an idea of present developments and probable future progress in design and manufacture. Treatment is given of the subjects of leakage, governing, etc., and the results of some hitherto unpublished researches on the impact of steam into the buckets and flow through them, and the erosion of metallic surfaces due to the impact of high velocity steam. Several side issues have been brought into the book in the shape of applied thermometry, and the subject of the specific heat of superheated steam, these being of particular importance in connection with steam turbine work.

The illustrations are almost entirely in the nature of line cuts, showing sections of various parts of the apparatus; diagrams showing the operation of steam, both in itself and in connection with nozzles and buckets; diagrams showing the relation between various features connected with the operation of the turbine, such as steam pressure, vacuum, consumption and power, and the expansion of steam. The efficiency of turbines is taken up in considerable detail, including the influence of the relative lengths and thicknesses of the vanes, and the effect of leakage. The text appears to be thoroughly well illustrated by the numerous cuts, and to cover the ground with considerable thoroughness. All of the principal commercial types of turbines are represented, and detailed descriptions given of each.

Elements of Gas Engine Design. By Sanford A. Moss, M. S., Ph. D. Size, 3¾ by 6 inches. Pages, 197. New York, 1906. D. Van Nostrand Company. Price, 50 cents.

This work is an attempt to present in condensed form all of the fundamental principles connected with the design of gas engines, giving a complete exposition of simply those elements which are of direct interest to the designer. No attempt was made to go into mathematical or constructional details. Most of the work treats of the Otto-cycle engine operating on the well-known four-stroke principle, and each discussion applies primarily to the single-acting trunk piston engine using this cycle. In some places, however, the discussion applies equally well to other cases. The subject is divided into fourteen chapters, of which the first five relate to the chemistry and physics of gas engines, and to various sorts of fuels used in their operation. The next chapter takes up the discussion of cylinder action, as affected by changes in exhaust and admission. Five chapters are devoted to compression, combustion, expansion, speed, power and efficiency, and methods of governing. The remainder of the work is concerned with cylinder action in two-cycle engines, details of design, and formulas and constants used by the designer. Four tables are given covering various subjects connected with the design and operation of the gas engine.

The Motor Boat and its Machinery Outfit. By Bruno Mueller. Size, 4¾ by 7¼ inches. Pages, 183.; illustrations, 126. Hanover, Germany, 1906. Dr. Max Jaenecke. Price, in paper covers, 2.40 marks; bound in cloth, 2.75 marks.

This little book is the sixth volume of a technical series being issued by the publisher, and is intended to bring before the reader in the most condensed manner possible the present condition of the motor boat industry in the line of design and operation. The illustrations consist of both line drawings and half tones, and cover in considerable detail both the boat and machinery.

The subject of the propeller comes up for attention, and turbine propellers, as well as reversible propellers, are given considerable space. The gearing for reversing the direction of operation of the propellers is also described, as well as the operation of propellers in tunnels extending above the water line. The engine is described in detail, not only the conventional types of motor boat engines being given, but also Diesel motors and producer gas outfits. Electric motors come up for brief consideration, and motor generator outfits are also described. The subject of the design and construction of the boat is given considerable attention, and is illustrated by both line drawings and half tones. The drawings cover both general arrangement plans and, in a few cases, the lines of the hull.

Boring and Turning Tapers. By Fred H. Colvin. Size, 5½ by 8 inches. Pages, 25; figures, 22. New York, 1906. The Derry-Collard Company. Price, in paper covers, 25 cents.

This is the second edition of No. 1 of the series of practical papers now being issued by the publishers, and is practical in every respect, showing the methods of turning tapers by means of offsetting the working centers of the lathe, as well as the operation of taper turning attachments with lathe centers in proper alinement. The subject of boring, and of cutting threads along taper lines, is given similar treatment, the whole outline of the work being of such a nature as to appeal directly to the man performing the various operations.

Transactions of the Institution of Naval Architects. Volume XLVIII, 1906. Edited by R. W. Dana, M. A., Secretary of the Institution. Size, 8½ by 11 inches. Pages, 228; folding plates, 40. London, 1906. Henry Sotheran & Company.

The scope of this work is well known, and all that is necessary in giving an outline of its character is to list the papers included and illustrated in its pages. These are:

The new Scouts of the British Navy.

Vessels Constructed for Service in Colonies and Protectorates.

Yacht Racing Measurement Rules and the International Conference.

The Subject of Motor Boats and Their Rating for Racing Purposes.

The Design and Construction of High Speed Motor Boats.

Gas Engines for Ship Propulsion.

The Efficiency of Surface Condensers.

Notes on the Freeboard Rules.

The Overhead Wire Cableway Applied to Shipbuilding.

The Introduction of Cranes in Shipyards.

Oil Tight Work in Ships of Light Construction.

Steam Yachts: Some Comparisons.

The work also includes lists of members, and minutes of the various business and professional meetings. It goes without saying that the illustrations are of exceptional scope and interest, and include a very large number of cuts, scattered over the forty plates. These are mostly in the nature of line drawings, but a few half tones are given of the various vessels described.

Bureau Veritas; thirty-seventh year; General List of Merchant Shipping. Two volumes: steamers, size, 10 by 7½ inches; pages, 1,685. Sailing vessels, size, 7¼ by 9¾ inches; pages 1,213. 1906. Paris, 8 Place de la Bourse; and London, 155 Fenchurch street, E. C. Price of the complete work, £3 3 s; steamers, £1 15s. Sailing vessels, £1 10s.

This work requires no introduction, it being the regular list of all merchant vessels of the various maritime powers, with the

usual particulars in the way of dimensions, powering, ownership, builders, etc. The books include also illustrations of the various types of vessels, together with tables showing the total tonnage and number of vessels of the various powers, as well as alphabetical lists of owners of fleets of vessels, with the respective tonnages. In addition to this, tables are given showing all the vessels under the several flags arranged in order of signal letters. At the end of the volume of Steamers is given a table showing all the dry-docks, patent slips, floating dry-docks and marine railways in all the ports of the world.

American Stationary Engineering. By W. E. Crane. Size, 5½ by 7¾ inches. Pages, 285; figures, 107. New York, 1906. The Derry-Collard Company. Price, $2.00.

The preface states that this book is intended to be a plain book of every day work about engines, boilers and their accessories. It makes no pretensions to scientific or mathematical treatment, all formulas used being placed in a simple form covered by a knowledge of arithmetic. No detailed description is given of such items as are treated in considerable detail in special books, these items being such as the indicator, slide valves, etc., but the work is expected to cover pretty thoroughly answers to all questions which are liable to come up in every day operation of a steam power plant. The illustrations are carefully drawn, and serve to explain the text in such a way as to make everything simple reading. At the end of the work will be found a considerable list of questions which would be likely to be asked at an examination for a license to operate steam machinery, and a number of tables have been collected from various sources, giving the principal characteristics of saturated steam, together with areas and circumferences of circles and areas of segments of a circle. The whole idea of the work is practical rather than theoretical, much of it being drawn from the personal experience of the author.

Internal Energy; A Method Proposed for the Calculation of Energy Stored Within Matter. By John V. V. Booraem, M. E. Size, 6 by 9¼ inches. Pages, 144; figures, 35. New York, 1906. McGraw Publishing Company. Price, $1.00.

The purpose of this work is to suggest a simple working hypothesis whereby the amount of chemical energy stored within a body may be estimated. It is based upon various lines of experimental data, the idea having originated several years ago from a mathematical study of the periodic curves of the atomic volumes and melting points. Through a combination of the methods of algebra and analytical geometry an attempt was made to discover why these curves were not solvable from the values of their ordinates. It was ascertained that to accomplish this purpose the determination of several constants was necessary, and also that as chemical energy was measured by heat, it was possible that chemical change could be measured by expansion through heat. On this basis the work was started, and includes considerations of expansion, as a measure of physical change. This is subdivided into general subjects of Cohesion versus Heat; the Individual Struggle between Atoms where Heat Measures Cohesion; the Energy of Atomic Pressure; Development of Chemical Energy; Units of Mass, Measure and Expansion; Chemical Volumes; Study of Stress and Volume by Groups; the Calory; Calculation of Latent Heat of Vaporization; the Origin of a Perfect Gas; the Melting Point; Specific Heat of Solutions; Latent Heat of Fusion; Methods of Calculating Energy of Cohesion; Liquidity.

SELECTED MARINE PATENTS.

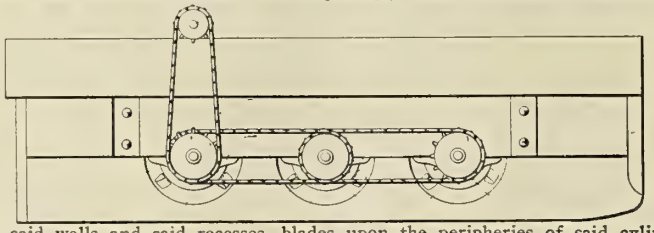
The publication in this column of a patent specification does not necessarily imply editorial commendation.

829,472. LIFE-SAVING DEVICE. OTTO HESSEL, BALTIMORE, MD.

Claim—1. A life-saving device comprising a frame raft, a series of air-tight compartments within the frame, of horizontal metal rods within the inner open frame parts, of a net slidably secured on the rods, of rope handles provided with cork floats secured and suspended from the exterior upper sides of the frame parts. Three claims.

829,058. BOAT. LUTHER J. BUTTERFIELD, MILWAUKEE, WIS., ASSIGNOR TO JAMES A. ALEXANDER, MILWAUKEE, WIS.

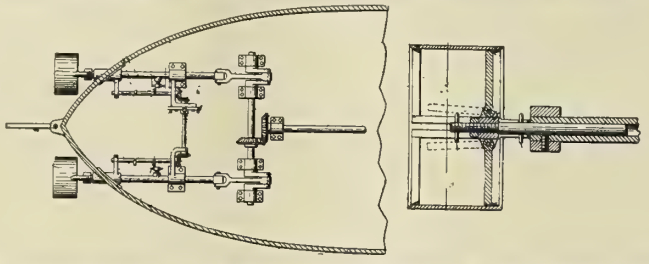
Claim.—3. A boat comprising a straight body or hull having tapered ends and a longitudinal series of wells, a keel upon said hull having recesses formed therein to aline with said wells, an overhanging deck upon said hull and formed with openings, hollow cylinders, journaled



in said wells and said recesses, blades upon the peripheries of said cylinders, a drive-shaft mounted upon said deck, rotary elements upon said shaft and shafts of said cylinders, and driving connections between said rotary elements. Three claims.

829,681. BOAT-PROPELLER. FRANK C. TAMBLING AND HARLAND CHARLES, EAGLE RIVER, WIS., ASSIGNORS OF ONE-THIRD TO CHARLES A. RUSCO, CHICAGO.

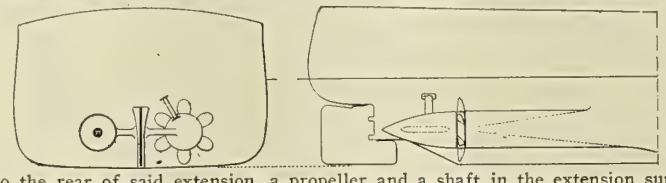
Claim.—1. In combination with a boat-hull, a tubular propeller-casing outside the hull; means by which it is mounted and reciprocated longitudinally on the hull; a gate-carrier mounted for longitudinal movement in



the propeller-casing; a gate hinged thereto adapted to swing wholly within the casing from position at which it obstructs the passage therethrough either way to unobstructing position, the propeller-casing having two longitudinally separated stops, and means for moving the gate-carrier to carry the gate at obstructing positions to one or the other stop at will. Three claims.

829,747. PROPELLER. ROBERT A. WORKMAN, PHILADELPHIA, PA., ASSIGNOR, BY DIRECT AND MESNE ASSIGNMENTS, TO LUTHER D. LOVEKIN, PHILADELPHIA.

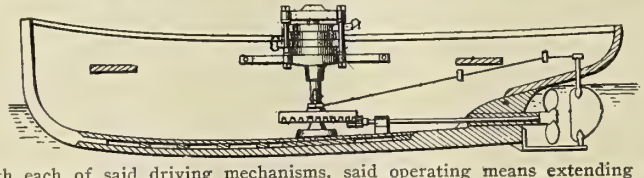
Claim.—6. The combination with the hull of a vessel of a tubular extension from the same, a substantially conical shell having its base adjacent



to the rear of said extension, a propeller and a shaft in the extension supporting said propeller between the end of said extension and the shell, there being a normally closed opening through the base of the shell adjacent to the shaft. Six claims.

829,782. BOAT. TIMOTHY HANLEY, BOSTON, MASS.

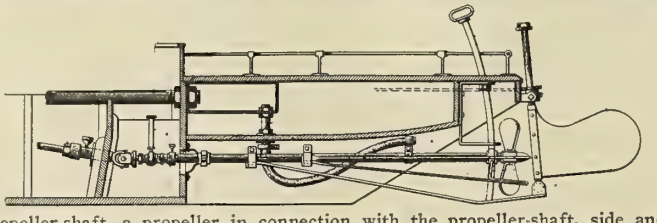
Claim.—1. The combination with a boat having a propeller-shaft, of a power-shaft geared thereto, duplicate driving mechanism arranged for cooperation with the power-shaft, and manually-operated means connected



with each of said driving mechanisms, said operating means extending in opposite directions from the driving mechanisms to permit operation of the propeller-shaft from either or both ends of the boat, and rudder-operating means revolubly supported by the power-shaft. Three claims.

830,908. MARINE VESSEL. JOHN W. MARTIN, SCOTTSBURG, IND.

Claim.— 3. A marine vessel having a propeller-shaft, a universal joint connecting said shaft with the driving-shaft, a tubular shaft encircling the



propeller-shaft, a propeller in connection with the propeller-shaft, side and bottom guard-rods, connected with the tubular shaft, a rudder-post having a rudder and connected with the bottom guard-rod, and means connected with the tubular shaft for vertically swinging the propeller shaft. Four claims.